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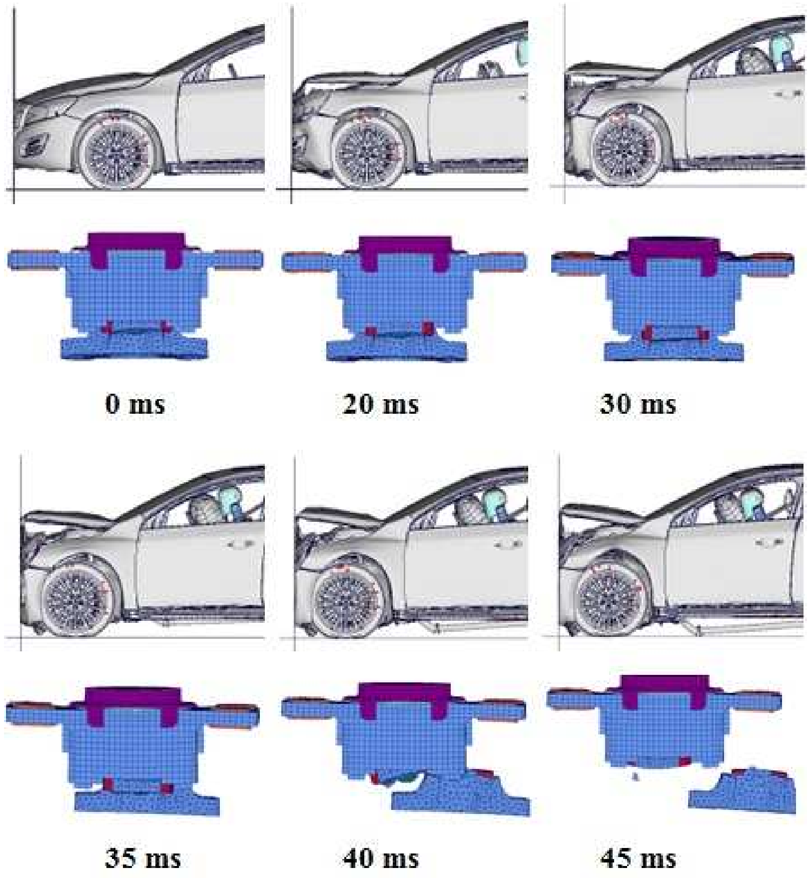

Modelling of Engine Suspension Components for Crash Simulations

Master's Thesis in the Master's programme Automotive Engineering

NILESH DHARWADKAR

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Department of Applied Mechanics
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CHALMERS UNIVERSITY OF TECHNOLOGY
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Cover:
Time sequence in a 35 mph Full Width Frontal Impact Crash Simulation showing the failure in the left hand side engine suspension mounts.

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ABSTRACT

Explicit finite element analysis (FEA) simulations are widely used in the product development of cars to evaluate their crash safety. Thus helping save substantial amount of resources required if only physical testing was to be used. At present there are several areas where simple modelling techniques are still used, primarily because of limited knowledge about the detailed behaviour of the system in a crash.

One such area is engine suspension components, e.g. engine and gearbox mounts. The objective of this thesis work is to develop new and improved finite element (FE) models for engine suspension mounts to be used in full vehicle crash simulations. These components play an important role in both high-speed crashes wherein failure occurs in the engine mount and in low speed crash where the dynamic properties of the engine mount are of interest in capturing the motion of the components in the engine bay. The new model should replace the usage of multi dimensional spring elements currently used in the existing models at Volvo Car Corporation (VCC).

The methodology used to achieve the objective was to perform component and sub assembly level tests in Fall Rig and correlate the test results by conducting simulations in the explicit solver LS-DYNA. Material models derived from the fracture curves are used to visualize the crack propagation in the aluminium casting.

A detailed FE model for engine mount has been developed from the observations made during correlation of Fall Rig tests. The new model was incorporated in the full vehicle crash simulations. The new model showed good agreement with the existing model for the engine mount and observations from actual crash tests.

Keywords: Crash Safety, Explicit FEA, LS DYNA, Crash Simulations, Engine Suspension Components, Physical Modelling.

Contents

Preface

This Master's Thesis Project is performed at Volvo Cars Safety Centre, Göteborg, Sweden at the frontal impact group. The work was performed during the period of September 2010 to February 2011.

The work aims at investigating techniques for developing detailed physical models for Engine Suspension Components in the front structure of the car that can be used in crash simulations at Volvo Cars Safety Centre. The thesis work was concentrated on developing Finite Element (FE) model for left hand side engine suspension mount that can be used in full vehicle crash simulations and capable of predicting the dynamics of the mount in a crash. The work ends with documenting recommendations for developing FE models for similar components in the front structure of the cars to be used in full vehicle crash simulations.

The work has been supervised by PhD Candidate Linus Wågström and Anders Sandahl, M.Sc from Volvo Car Corporation and examined by Asst. Prof. Karin Brolin, Head of Vehicle Safety Division, Chalmers University of Technology.

Göteborg February 2011

Nilesh Dharwadkar / Krishna Prashant Adivi

Acknowledgements

This work is an internal development project carried out at Volvo Cars Safety Centre, Göteborg, Sweden with frontal impact group. We are thankful to all the people that have followed and helped us during the project.

We are very thankful to our supervisors PhD Candidate Linus Wågström and Anders Sandahl M.Sc for providing us with an opportunity to work on this project. We also wish to express our gratitude to them for providing us the guidance and constant feedback during the project and sharing their invaluable knowledge and experience within Vehicle Crash Simulations. We would like to thank PhD Johan Jergeus at the Volvo Cars Safety Centre for sharing his knowledge and experience in material modelling for crash simulations. We thank Oscar J. Centeno G from Xdin AB for sharing experience from his thesis work. We would also like to thank Reino Frykberg in helping us build the test rigs for physical testing. We express our gratitude to all the members in the Frontal Impact group at Volvo Cars Safety Centre for making our stay at their department pleasurable.

We would like to show our appreciation to Asst. Prof. Karin Brolin for providing feedbacks on our report and examining our work.

We express our pleasure in thanking our friends Govind, Lobhas, Vamsi, Mukul, Sandeep K, Nirup, Bharat and our 'Desi Automotives' at Chalmers for their pleasureful company and support during our stay in Sweden.

Last but never the least we wish to express our special thanks to our parents and siblings for supporting our decisions and dreams. It is their faith in us and constant guidance that has helped us to achieve success in our master's studies in Sweden. It is their love and encouragement that motivates us to achieve our goals in life.

Thank you all!!

Göteborg, February 2011.

Nilesh Dharwadkar / Krishna Prashant Adivi

NOMENCLATURE

Greek Letters

Δt	Time step for FE simulations
ρ	Mass density
ω_{max}	Natural frequency

Roman upper case letters

C	Damper matrix
C_{ij}	Rubber material parameter
E	Young's modulus
F	External force
I	Strain invariants
K	Stiffness matrix
M	Mass matrix
U	Nodal displacement vector
$\dot{U}$	Time derivative(first order) of nodal displacement
W	Strain energy density function

Roman lower case letters

a_0	Arbitrary constants
c_e	Speed of sound in the material
d	Material coefficients
l_e	Minimum element length
v	Poisson's ratio

Abbreviations

CAD	Computer aided design
CAE	Computer aided engineering
FE	Finite element
FRELIM	Failure risk for element elimination
IGES	Initial graphics exchange specification
LHS	Left hand side
MF GenYld	MATFEM Generalised yield locus
NHTSA	National highway traffic and safety administration

1 Introduction

1.1 Background

In automotive product development there has been an ever increase in the application of Computer Aided Engineering (CAE) techniques for simulation of crash event, particularly due to the availability of high computing machines and parallel computing techniques. The current capabilities in structural crashworthiness simulation through CAE analysis is an important reason for increase in safety standards. The two most important reasons for use of crash simulations are to know the effect of impact on the vehicle structure and to analyse the safety of the occupants. These simulations offer today reasonably accurate results and save significant amount of resources that would have been otherwise used in physical testing. Hence use of CAE techniques result in an efficient product development cycle.

Though there has been tremendous increase in the use of CAE tools within automotive crash safety, still the best is yet to come and research is very active in investigating the possible use of more physics based models. In current CAE models several areas are still handled in a much simplified way, e.g., by use of multidimensional spring elements instead of a detailed physical model. One such component is engine suspension mounts.

The engine suspension mounts are used to suspend the engine, gearbox and the driveline with respect to the body structure. The main function of the mounts is to form the connection between the engine and the body structure, to isolate the vibrations going into the body during dynamic scenarios and avoid transmission of noise.

The current level of details being captured in full vehicle crash simulation models has not allowed detailed modelling of engine suspension mounts with solid elements. The main reason for this is time required in developing reliable good quality meshed models for cast aluminium and rubber, accurate material models for rubber and cast aluminium, and lack of knowledge of the kinematics of the engine mounts under crash.

The important areas of focus of this master thesis work have been to develop an improved Finite Element (FE) model for the Left Hand Side (LHS) engine suspension mount to be used in full vehicle crash simulations that is capable to capture the physical behaviour of the mount and failures during the crash. To achieve this, the software for non-linear dynamic analysis of structures in three dimensions crash LS DYNA [8] was used.

The model for LHS Engine Suspension mount currently used in full vehicle crash simulations at Volvo Cars Safety Centre is simplified with use of shell elements, rigid material models and multidimensional spring elements. This model is tuned by varying the force properties in the spring element based on the judgement of the CAE engineer and observations from the crash tests. Thus the current model is not capable in capturing the behaviour of the engine mount's rubber bushing, hydraulic fluid and failure within aluminium casting.

1.2 Objective

The objective of the thesis work is to develop an improved FE model for the LHS Engine Suspension Mount that captures the construction details necessary to simulate the actual behaviour of the mount under crash. With a detailed physical FE model built with solid elements and advanced material models, it is possible to capture the nonlinear behaviour of the rubber bushing, bending in the bolts and crack propagation within aluminium silicon casting.

The new FE model is validated against the test data (acceleration pulse) recorded during physical testing of the engine mount in the Fall Rig tests. The material models for rubber and cast aluminium are tuned and new FE model for the engine mount were then incorporated into full vehicle crash simulations. The performance of the new FE model was validated by comparing observations made from actual crash tests.

1.3 Limitations

The thesis work is concentrated on developing a physical FE model for LHS engine suspension mount. The modelling technique can be extended to other engine suspension mounts.

However, the modelling technique and material models used for modelling engine suspension mounts have not been tested to be used for modelling components with similar materials or functions. Thus additional tests and investigations must be performed to check if the modelling techniques and material models from this thesis work can be extended for similar components.

2 Theoretical Frame of Reference

This chapter discusses the history, functional behaviour and construction of engine and gearbox mounts.

2.1 History of Engine Mounts

Engine and gearbox mounts connect the heavier parts of the vehicle, such as engine and gearbox to its body or chassis. The purpose of these components is to support the engine and isolate the vibrations from the engine and road excitations.

In the earlier days the construction of these types of mounts was simple where rubber was bonded between metal casings and the engine was supported on to the body. Rubber is used in vehicles since the early 19th century, Britannica [15]. Walter Chrysler was the inventor of rubber mounts and encouraged to use them for the isolation of vibrations. It was first used in Ford's Plymouth model where the engine was supported on three points, resting on the rubber mount. Noise and vibrations have reduced although there were still some when the engine is in idle, but under load the rubber mounts have shown significant reduction of noise. Now rubber systems are an integral part of the vehicle.

These mounts had low elastic stiffness and low damping characteristics since it was made of mostly rubber and they were only suitable to isolate high frequency vibrations exerted by the engine. The main disadvantage of these conventional engine mount is they exhibit poor performance to road excitation, also called engine bounce, where the vibrations caused are of higher amplitude, Dukkipati [14]. The most common engine suspension components used are engine mounts, gearbox mounts and the torque restrictor. Usually there are two engine mounts, one connects gearbox to body and the other engine to the body. They are normally arranged to provide a degree of flexibility in the horizontal longitudinal, horizontal lateral, and vertical axis of rotation. A torque restrictor is used to limit the torque movement of the engine in most modern vehicles where the engine is mounted transversely, because the engine adopts a rolling movement when transmitting loads.

In the recent years, significant improvement to the conventional engine mounts have occurred that can support the engine and also have the capability to damp engine vibrations as well as road excitations enhancing the dynamic stiffness of the rubber bushing. Thus to meet the design requirements it is necessary that these mounts are made stiff to support the engine and have damping amplitude dependent of excitation.

For most powertrain installation, Volvo Car Corporation uses a pendulum type engine mounting system wherein the engine is mounted in transverse direction. There are a total of three engine mounts used to connect the powertrain to the body structure. These three mounts are shown in figure 2.1

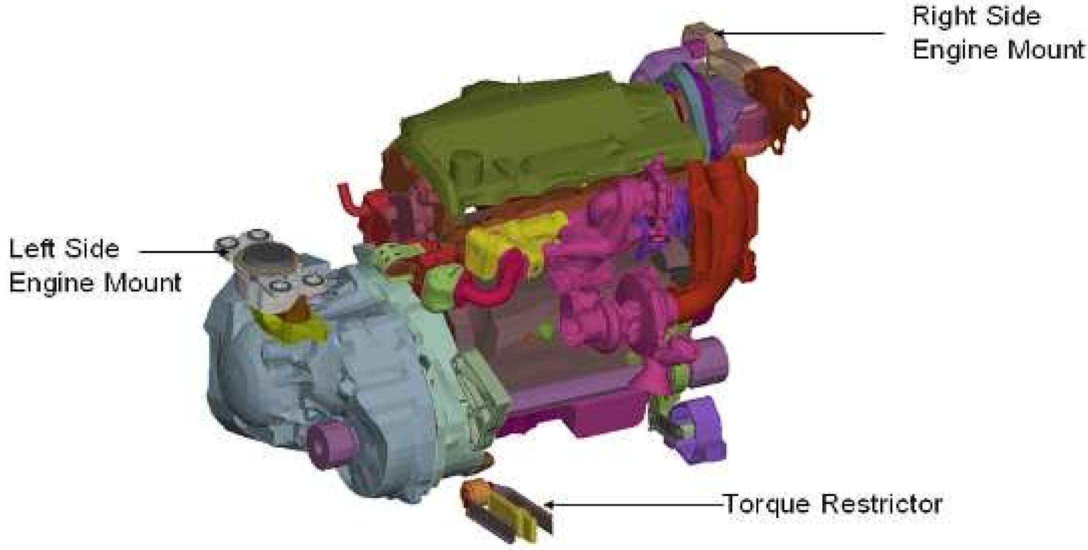

Figure 2.1 Positions of Engine Suspension Components in Volvo Car Corporation Transmission Architecture

2.1.1 Construction

The gearbox mount referred to as 'LHS Engine Suspension Mount', connects the gearbox to the body side members of the vehicle, see Figure 2.2. This engine mount is manufactured by Trelleborg Automotive, Germany and is used in most of the production vehicles at Volvo Car Corporation. It is situated in the left side of the engine bay area parallel to the longitudinal axis of the vehicle along the direction of motion, see Figure 2.1.

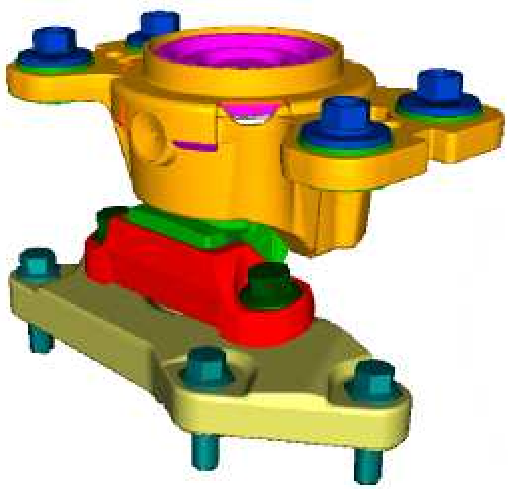

Figure 2.2 Arial View of LHS Engine Suspension Mount

The upper and lower parts of the gearbox mount are made of an Aluminium Silicon (AlSi12Fe) alloy casting and are connected together by a screw joint. Vulcanized rubber acting as an elastomeric spring is bound to the inner wall of the upper gearbox mount. This rubber bushing absorbs the static loads through vertical axis of the mount. There are two chambers within the hydro-mount. The upper chamber and the lower chamber are separated by an inertial track and an orifice plate combined to form a hydraulic circuit. This circuit plate is made of plastic. The inertial track is a helical channel and the orifice plate works as a decoupler containing numerous transfer holes. These two chambers are filled with a hydraulic fluid made of water and glycol in equal proportions.

This flow of the fluid within these two chambers is controlled by the plastic circuit depending on the amplitude of vibration. When the engine is idle it produces vibrations of small amplitude and high frequency range, then the fluid flows through the decoupler and restricts the flow through the inertial track. This is similar to a conventional rubber mount. When the vehicle is subjected to road excitations it causes large amplitude vibrations of low frequency, in this case the flow of the fluid is through the inertial track. Detailed characteristics of the hydro-mounts are given by Colgate [2], Singh [3], Heinz [4], Dukkipati [14] and Renault [5].

2.1.2 Engine Mount Role in Crash

As discussed in the section 2.1.1 the main function of the engine mount is to isolate the vibrations caused by the engine and road surface. The behaviour of the engine mount and the rubber bushing is vital in frontal crashes. In high speed impacts, failure in the engine mounts occur, while in low speed impacts, the dynamic behaviour of the engine mount is of prime interest. This section discusses the role of the engine mounts in frontal impacts.

The front structure in a Volvo car is designed with an objective to distribute energy over the entire body in the event of a crash and absorb the collision forces in an effective and controlled manner. The engine also contributes to effective deformation with its space saving transverse installation.

In the event of frontal collision, the relation between the passenger compartment deceleration and interaction forces is complex due to the unique properties of different components with respect to geometry, inertia and stiffness. In case of the engine, there is a specific distance by which the engine is displaced before it contacts the firewall. The engine suspension mounts allow engine displacement relative to the body structure.

The engine has substantial mass and inertia, such that a frontal impact at high speed can cause the engine to move forward in its mounts, until it impacts the barrier, Boughton [11]. On impacting the barrier the engine is decelerated and pushed rearward to make contact with the firewall. The forces in the crash are sufficiently high so as to cause the failure in the engine suspension mounts that results in the heavy parts in the powertrain being disconnected and the body structure and being made to absorb the energy in the crash in a controlled and efficient way. In case the engine mounts don’t fail in the crash, the engine may be forced to move further against the firewall resulting in intrusions in the passenger compartment.

In case of low speed impacts, the impact energy is not high enough to disconnect the engine. However, the dynamic motion of the engine is important. This motion depends on the characteristics of the engine suspension mount that support the engine with respect to the body structure.

To summarize, in a high speed impact failure in the engine mounts is essential whereas in a low speed impact the engine mount remains intact and its dynamic properties are important.

2.2 Material Properties

The engine mount being discussed in this thesis has Aluminium Silicon Alloy and Rubber as its major material. This section describes the properties of these materials.

2.2.1 Aluminium Silicon Alloys

Aluminium alloys are widely used in the automotive industry, especially in the engine bay area. This is mainly done to reduce the weight of the complete vehicle. Most of the components are made up of Aluminium Silicon alloys. Due to the complexity of the components it is necessary that these materials should possess good cast-ability and resistance to corrosion and are made of A 413.0 which corresponds to eutectic composition of 12% Si by weight [7].

The mechanical properties of these components are strongly dependent on the microstructure of the alloys. These properties can be modified by controlling cooling and solidification rates of the alloy. The rate of cooling influences the size, shape and interdendritic spacing of the micro constituents. The rate of cooling depends on the pressure maintained during the casting process. The casting method that is used depends on the amount of silicon in the alloy. Therefore the rate of cooling is also controlled by the amount of silicon. The percentage of silicon by weight also influences the mechanical properties of the aluminium alloy. Increasing the silicon percentage increases the strength and ductility of the alloy. The casting process to be chosen, depends on the amount of silicon and AlSi eutectic system

The following figure show the microstructure of 12% Si Aluminium Alloy

12% Si

Figure 2.3 Microstructure of cast AlSi12 alloy

During casting process, when the liquid transforms into solid there is a possibility that the volume decreases. This shrinkage results in formation of porosity in the alloy which can degrade its mechanical properties. Porosity is common in Al-Si castings. A pore in the alloy is a stress concentrator and when subjected to any external load, it can lead to micro crack initiation and propagation [7].

2.2.2 Rubber Properties

This section discusses the properties of rubber used in engine suspension components. Rubber in its natural state is too soft to be used as a vibration isolator. To improve the mechanical properties i.e., strength and elasticity vulcanization of rubber is done. Vulcanization is an irreversible chemical process discovered by Charles Goodyear in 1839 to make rubber tires [8]. In this process a polymer usually rubber is heated by adding a curing agent to form cross-linked chains between molecules of rubber increasing its elasticity and stability. This vulcanized product of rubber can also be called as elastomers. The most common curing agent is Sulphur. The number of Sulphur atoms to the cross links determines the mechanical properties of the elastomer i.e., if the cross links contain more sulphur atoms the elastomer has very good flexibility while it exhibits poor heat resistance. Elastomers are ten times stronger and more rigid than natural rubber [9].

The capacity to recover from large deformations is one of the distinguishing mechanical properties of rubber. Certain rubber compounds can recover from nominal strains of up to 600%. What is particular about this behaviour is the nonlinear stress-strain relationship encountered in such deformation. It is generally characterized by initial softening, then sudden stiffening as the material approaches its elongation limit [9].

Rubber materials are load rate dependent. The strain rate has a major effect on the stiffness, which increase dramatically in rapid process. This behaviour can partly be described as viscoelastic. The major part of the relaxation occurs in a very short time. The relation between the shear modulus and the bulk modulus is large, the bulk modulus is usually 1000-2000 times higher than the shear modulus. This makes rubber nearly incompressible. Thus, in many cases the approximation of incompressibility is quite appropriate.

3 FE Model of Engine Suspension Mount

This chapter describes the development of the new FE model for the LHS engine suspension mount based on the requirements generated while setting the objective of the thesis work.

3.1 FE Model Development

The engine suspension mounts are developed and manufactured for Volvo Car Corporation by Trelleborg Automotive, Germany. The CAD model for the mount is developed in design program CATIA V5. The data required for FE modelling was acquired in Initial Graphical Exchange Specification (IGES) format that is supported in all the CAE pre – processing software.

The pre – processing software used in the thesis work was ANSA, available from Beta CAE Systems S.A. ANSA is an advanced multidisciplinary CAE pre-processing tool that provides all the necessary functionality for full-model build up for crash analysis, from CAD data to ready-to-run solver input file, in a single integrated environment.

The following steps were carried out in ANSA:

- Importing IGES file into ANSA environment.
- Cleaning of geometry and preparing the file for mesh generation.
- Generating appropriate mesh for individual components.
- Assembling meshed components and performs quality checks for mesh generated.
- Assignment of appropriate material models and material properties for different components.
- Define contacts between different parts in the assembly based on the physical behaviour of the component.
- Define necessary boundary conditions like displacements, velocities, forces etc.
- Solve the FE model in LS DYNA.

3.2 Mesh Generation

This section describes the strategy used in developing the mesh for the new FE model for the engine mount.

3.2.1 Hex Vs Tetra Meshing

Tetra meshing is a fast approach for generating solid mesh for a component. Automatic and 2D (Tria) to 3D (Tetra) meshing techniques can be used to generate solid mesh in a short time compared to hex mesh. These techniques also help in capturing the geometry of the solid component more accurately compared to hex mesh. However tetra mesh generally results in large number of elements and nodes in meshing the same component as compared to hex mesh. Thus resulting in a higher solution time. A comparison was performed between tetra and a hex mesh to mesh the aluminium casting. The results for this study are documented in the Chapter 5.

The hex mesh is generated semi – automatically or manually and requires more time and skill. However, hex mesh creates less number of elements and nodes to mesh a solid component compared to a tetra mesh for a given element length. Hence resulting in lower solution times. With a hex mesh it is difficult to capture the geometry of complicated solid shapes and hence approximations through geometry cleanup are often made.

3.2.2 Selection of Mesh Parameters

The most critical mesh parameter for crash simulations is element length. The element length for the mesh is selected based on the time step being used in the crash simulations so as to have stable mesh with minimum mass scaling.

The element lengths are mentioned in Table (3.1) below:

Table 3.1

Sl. No	Material	Element Length (mm)	Element Type
1.	Aluminium Casting (AlSi12Fe)	5.5	Underintegrated Hex Elements
2.	Rubber (52 Shore)	4	Underintegrated Tetra Elements
3.	Plastic	3	Underintegrated Hex Elements

3.2.3 FE Model Development

Figure 3.1 shows the final FE model for the LHS engine suspension mount. Since the geometry and construction of the engine suspension mount is complex, approximations had to be made in suitable areas in developing the FE model without affecting the physical behaviour of the mount.

Though a fine mesh for all the components in the mount would produce a result with higher degree of accuracy, it was necessary to make approximations and not decrease the element size beyond a critical value to have stable model with minimum mass scaling with reasonable computational time. The hydraulic fluid in the mount is not modelled since it has no significance in crash loading of the engine mount as explained in section 2.1.1 in Chapter 2. The aluminium casting has been modelled to capture its shape to the maximum extent. However the geometry has been cleaned to remove unnecessary holes and fillets to decrease the level of details and complexity. The rubber bushing is modelled using tetra elements since modelling it in hex elements was difficult due to its complex shape. Some parts in the rubber bushing were not modelled as it resulted in large distortion in the elements during simulations causing negative volume and stability problems. The vulcanised rubber glued to the lower gearbox mount was approximated and modelled as aluminium casting due to its small thickness. The rubber cap on the top aluminium casting that forms the boundary for the hydraulic fluid was modelled as aluminium instead of rubber to provide better visualization of the mount during post processing. The plastic hydraulic circuit in the top aluminium casting was modelled in a much simpler form. The significance of the plastic circuit is only to add the mass to the FE model.

The aluminium casting was modelled using eight node solid brick element as well as four node solid tetrahedral element to form two versions of the FE model. The rubber bushing was modelled using four node solid tetrahedral elements. The top aluminium flange and the rubber boundary were modelled using four node quadratic shell elements. The screw in the mount is modelled using two node beam element. Since LS DYNA does not consider the cross section that was inputted while defining the beam element, a cylindrical casing was created around the beam elements representing the bolt. The cylindrical casing allowed for the physical dimension of the bolt and the contact between the bolt and the aluminium casting working properly. The cylindrical casing for the bolt was modelled using 4 node quadratic shell element. The dimension of the cylindrical casing was kept smaller than that in the CAD model to compensate for the thickness of the shell elements. Thus avoiding any contact penetrations between shell elements and also not affecting the behaviour of the model.

Solid elements for aluminium casting are modelled as under-integrated since they were less costly and were able to replicate the behaviour in the aluminium casting that does not undergo big deformations. The solid elements in the rubber bushing were modelled as fully integrated since this element formulation performs better with high deformations, however is more costly than single point integration. The shell elements and beams were modelled as fully integrated shells with Belytschko - Tsay formulations and Huges – Liu beam with cross section integration respectively.

The meshed model for the LHS engine suspension model is seen in figure 3.1 below:

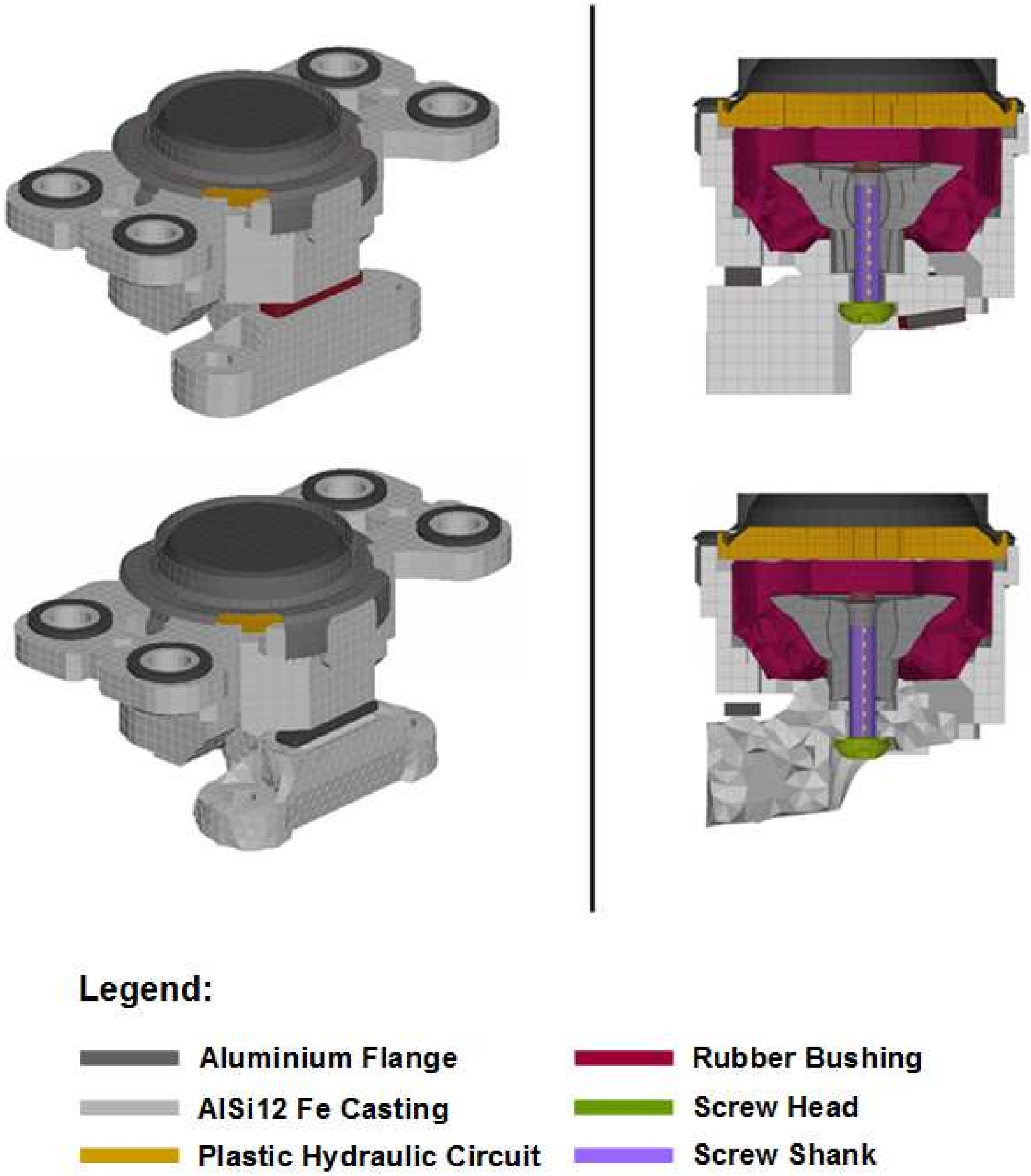

Figure 3.1 Arial and cross sectional views of FE model of LHS Engine Suspension Mount. The top row shows the lower gearbox mount meshed with hex elements whereas the second row shows the lower gearbox mount meshed in tetrahedral elements. Both models were studied in full vehicle crash simulations.

3.3 Material Model Selection & Assignment of Material Properties

The material model used for the aluminium casting is a material card 48, MAT_USER_DEFINED_MATERIAL_MODEL 48. This material card is generated based on the CrachFEM material data. Additional details for this material card are available in APPENDIX B.

The rubber bushing with 52 shore A rubber is defined using material card 77, MAT_HYPERELASTIC_RUBBER. The hyper-elastic material model is based on Yeoh model. Additional details for this material model are available in APPENDIX B.

3.4 Contact Definitions

In the complete engine suspension mount global contacts were used for all the metal to metal contacts. The global contacts allow for a friction of 0.2. Contact between the rubber bushing and the aluminium casting surface was created by implementing the contact option TIED_NODES_TO_SURFACE_OFFSET available in LS DYNA. This contact ties the nodes on one surface to the surface of the other part. An optional thickness of 30 mm was defined so as to capture the nodes that are far away from the contact surface. This was necessary due to coarse mesh for the aluminium casting.

The top rubber cap in the mount is clamped between the aluminium cap and the upper aluminium casting. This contact was simplified by merging the nodes of the upper rubber cap to the plastic circuit cap and using nodal rigid body connections between the upper rubber cap and aluminium cap.

4 Physical Testing

This chapter describes the component and sub – assembly testing carried out in this thesis work and results obtained from the tests.

4.1 Background to Testing

The objective of the component testing is to validate the material model used for AlSi12Fe casting under crash loading and validate the simulations performed in LS-DYNA. The motivation to carry out component testing is to tune the material cards for aluminium silicon (AlSi12Fe) Casting.

The tuned material card for AlSi12 casting and rubber from component testing phase were used in the new FE model for the engine mount. In second phase, drop tests were performed on the engine mount assembly. The test results were correlated with LS – DYNA simulations to validate the new FE model for engine mount.

4.2 Physical Testing Plan

The physical testing in the thesis work was carried in two phases. In the first phase the lower gear box mount and in the second phase the complete LHS engine suspension mount assembly were tested in the Fall Rig. Figure 4.1 shows the flowchart explaining the linkage between the physical testing and other important stages in the thesis work.

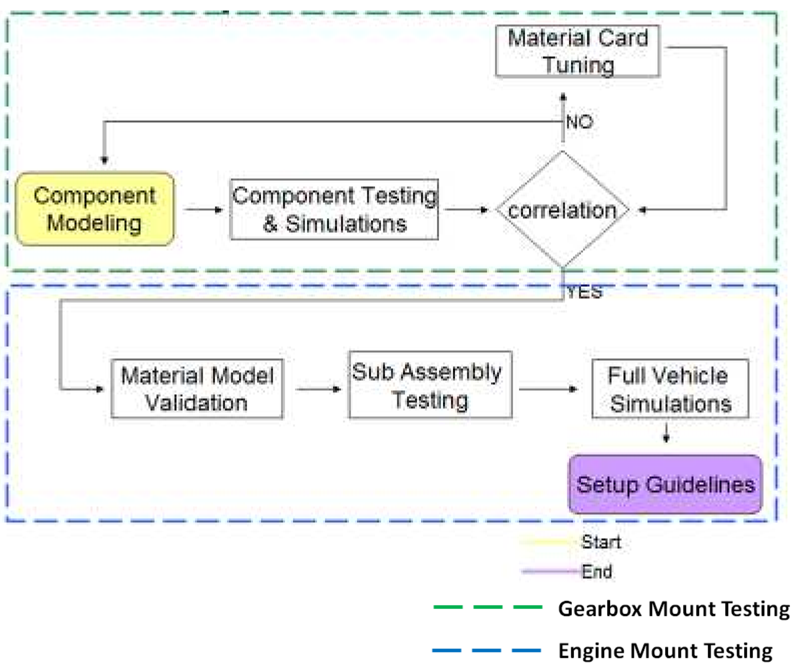

Figure 4.1 Flow chart picturing the Physical Testing Phases

4.3 Testing

4.3.1 Gearbox Mount Testing

The lower gear box mount in the engine mount assembly is shown in figure below:

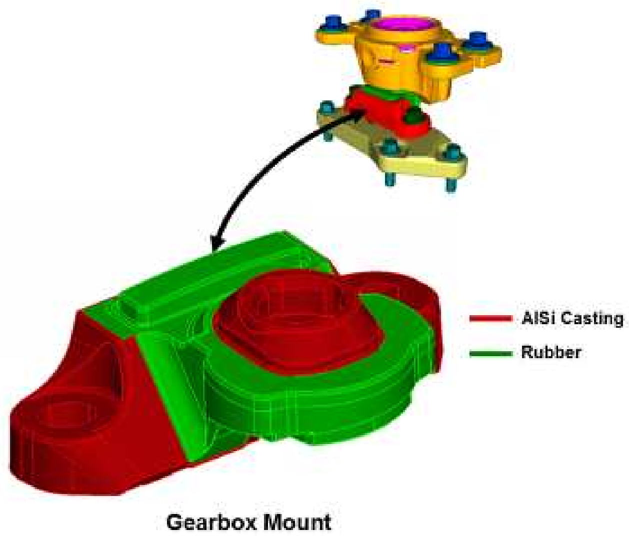

Figure 4.2 Gearbox Mount Location in Engine Mount Assembly

The gearbox mount is as AlSi12Fe casting with vulcanized rubber of shore value 52 glued to it. The shore value of rubber was obtained from the manufacturer of the engine suspension mount. The mount is bolted to the gearbox through an aluminium bracket. The rubber damps metal to metal contact between the gearbox mount and the upper engine mount assembly during dynamic movements in the engine bay and is not expected to have a significant role in absorbing crash loading in frontal impacts.

The test setup for the gearbox mount testing is show in the figure 4.3.

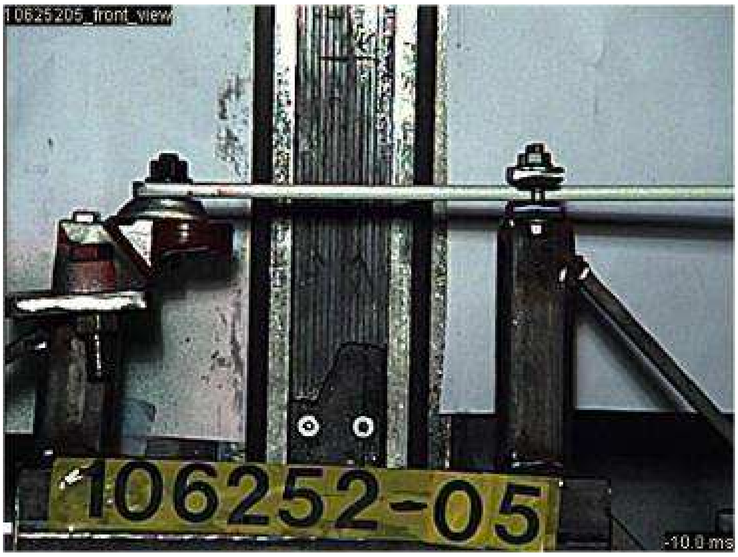

Figure 4.3 Phase One Test Setup for Gearbox Mount Testing

The test setup was designed to load the gearbox mount indirectly by impact the drop weight hammer on 8 mm thick mild steel plate. The test rig was made of mild steel plates of 6 mm and 12 mm thickness welded together and clamped to the bench of the Fall Rig. The test rig was designed to have high rigidity to avoid flexing during testing. High rigidity of the test rig also ensured simplicity in modelling the weld connections between the plates in FE model used for correlating the test results and good repeatability between different tests.

4.3.1.1 Test Configurations for Gearbox Mount

In phase one testing of the lower gearbox mount, the drop mass and initial velocities were determined with an aim of causing a fracture in the gearbox mount. The values for drop mass and initial velocity were approximately estimated based on basic mechanical design calculations and the limitations in the Fall Rig.

Based on the approximate calculations, the tests on the lower gearbox mount were performed with drop weight of 23Kg from a height of 7 m. Total five tests were performed. The first two tests were used to calibrate the instruments and the test setup and last three tests were used to perform correlations with the LS DYNA simulations. The gearbox mount fractured in all the tests.

4.3.2 Complete Engine Mount Physical Testing

The testing for the complete engine mount was carried out at the Fall Rig in Volvo Cars Safety Centre. The objective of the testing was to evaluate the behaviour of the engine mount under dynamic loading and to check the influence of hydraulic fluid on the response from the engine mount.

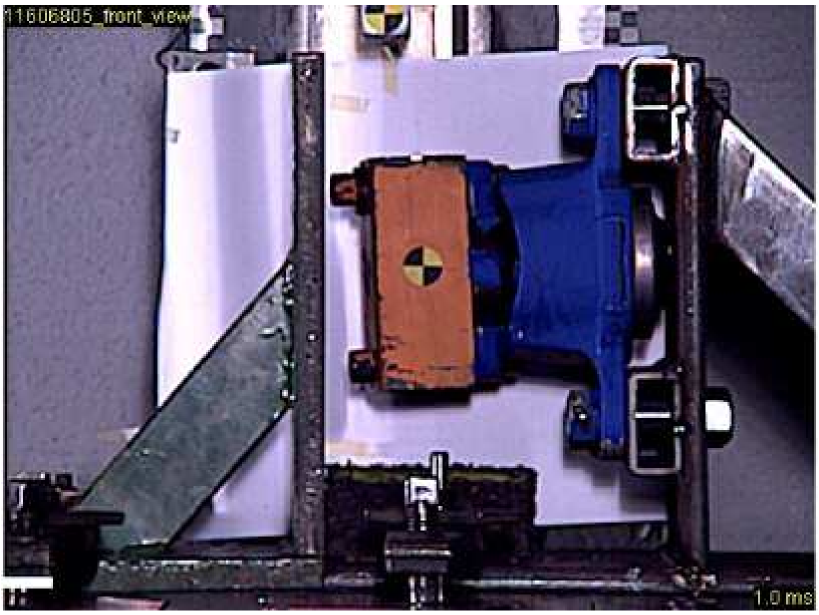

Figure 4.4 Phase 2Test setup showing complete engine mount.

The test rig was made of 6mm and 15mm thick mild steel plates welded together and then clamped to the bench of the fall rig. The complete engine mount was bolted vertically on to the test rig as shown in figure 4.4. The drop weight was made to impact on to a rigid block of dimension 60x60x150mm bolted to the lower gearbox mount.

A total of 11 tests were performed in the phase 2 testing, out of them few initial tests were conducted for the calibration of instruments and also to ensure the performance of the test rig and the other tests were to evaluate the behaviour of the rubber bushing and to analyse the effect of hydraulic fluid on the response of the engine mount at different speeds and impact masses. From the initials test runs it was observed that the rubber bushing was very flexible causing excessive rotation of the lower gearbox mount on impact of the drop weight. To constrain this rotation a support plate was introduced to the test rig so as to achieve vertical motion of the lower gearbox mount relative to the upper body of the engine mount.

4.3.2.1 Test Configurations Engine Mount

Tests were performed on the complete engine mount assembly with and without hydraulic fluid. Different configurations of initial speed and impact mass were tried. Based on approximate calculations and observations from trial runs, the initial velocity was set to 4.4m/s and their corresponding mass was 38kg.

In the figure 4.4 it can be observed that there is less clearance between the support plate and the bolt used to connect the rigid block to the aluminium mount. Thus during the test as the lower gearbox mount flex's in the rubber bushing, the bolt makes contact with the support plate resulting in significant friction. The tests were performed by lubricating the surface of the support plate to minimise the friction. However, it was critical to account for this friction while correlating the LS DYNA simulations with tests in phase two.

4.3.3 Instrumentation

Test measurements were obtained using force transducer, accelerometers and high speed camera. The force transducer used in these tests measures forces in vertical direction and has a capacity to measure up to 110kN with a resolution of 16bit that can collect 9600 data points every $6.25\ 10^{-5}$ms. There were two accelerometers to measure the acceleration in the impact hammer. Both the accelerometers and the force transducer are placed on top of the impactor. The high speed camera was used to film the dynamic behaviour of the engine suspension mount and make visual comparison with the FE simulations. The high speed camera used in this testing has the ability to record 1000 frames per second. Two cameras were used, one to film the front view of the test rig and the other in the lateral direction. An optical trigger was used that activates the instruments and starts recording the data.

The raw data obtained from the tests was processed using software Hypergraph 10 from Altair Engineering to obtain desired plots necessary for analysis.

4.4 Results

4.4.1 Results of Phase One Testing

This section presents the results for the first phase of testing done with the lower gearbox mount. The test for gearbox mount were performed three times. The results from these tests are represented in figure 4.5, figure 4.6 and figure 4.7.

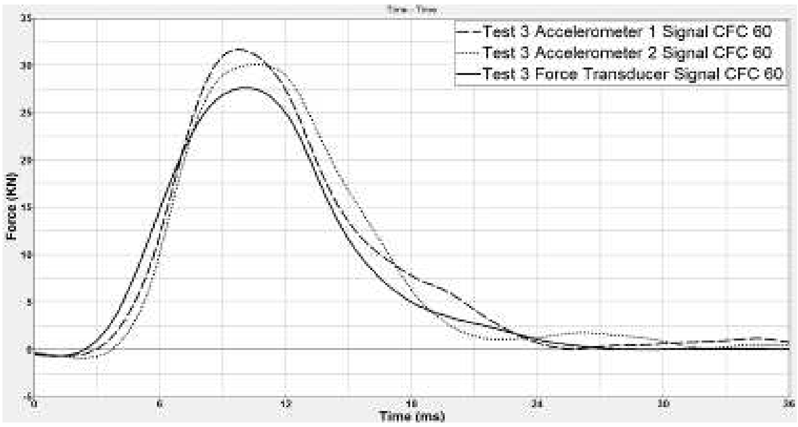

Figure 4.5 Force Vs Time plot for Phase One Test 3.

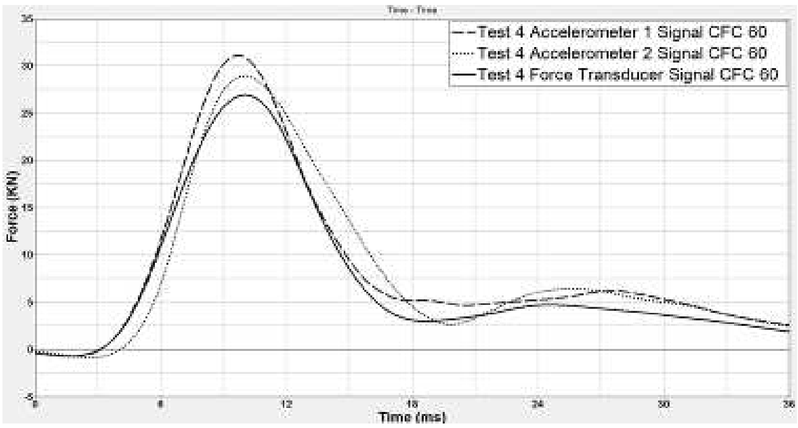

Figure 4.6 Force Vs Time plot for Phase One Test 4.

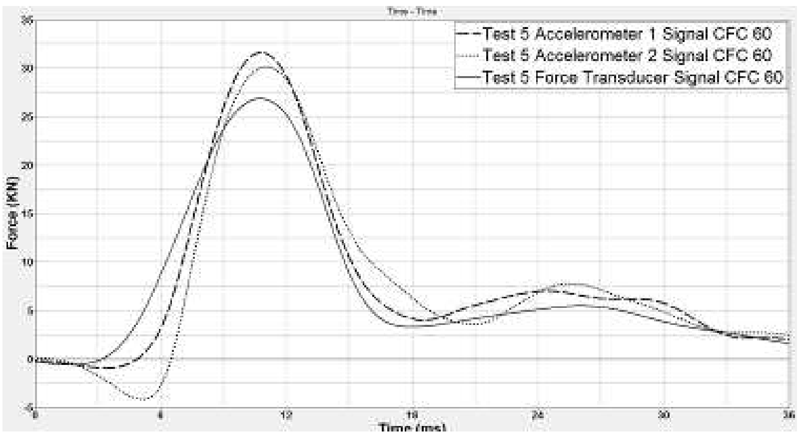

Figure 4.7 Force Vs Time plot for Phase One Test 5.

4.4.2 Results from Phase Two Testing

4.4.2.1 Complete engine mount with hydraulic fluid

The results from the tests on the complete engine mount assembly with the hydraulic fluid at speed of 4.4m/s and impact mass 38kg are presented below in the figures 4.8, 4.9 and 4.10.

Figure 4.8 Force Vs Time phase 2 Test 3.

Figure 4.9 Force Vs Time phase 2 Test 4.

Figure 4.10 Force Vs Time phase 2 Test 5.

4.4.2.2 Complete engine mount without hydraulic fluid

The results from the tests on the complete engine mount without the hydraulic fluid at speed of 4.4m/s and impact mass 38kg are presented below in the figures 4.11 and 4.12.

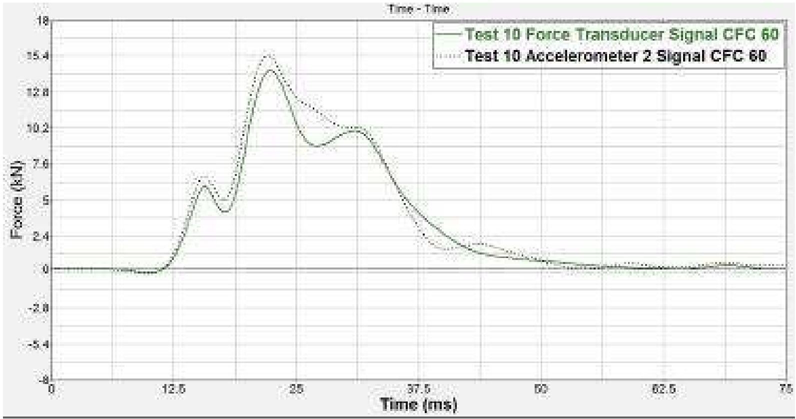

Figure 4.11 Force Vs Time phase 2 Test 10.

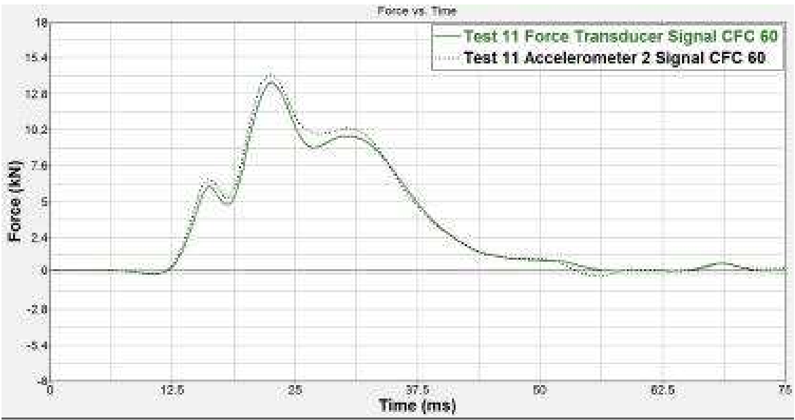

Figure 4.12 Force Vs Time phase 2 Test 11.

4.5 Analysis & Discussion

Results from the testing , it can be inferred from the figures 4.5, 4.6 and 4.7 the signal from all the three sources viz, two accelerometers and force transducer, are consistent in all three tests with no significant difference. The peak value of force in all the test was ~ 28KN. Any differences observed can either be due to the non-uniformities in the aluminium casting (AlSi12Fe) or from dynamic effects during testing such as friction in the test rig.

Analysing the plots from the tests performed in phase two, engine mounts with the hydraulic fluid it is observed that the maximum force measured in all the tests is ~ 14.1kN (refer figure 4.8 to 4.10). Similar is the case in tests on engine mounts without hydraulic fluid where the measured maximum force is 14.1kN (refer figure 4.11 to 4.12).

Studying the results can confirm that the presence of hydraulic fluid inside the engine mount has no effect on the response from engine mount under impact loading conditions. With respect to section 2.1 it can be verified that the major function of hydraulic fluid is to prevent transmission of vibrations from engine to the vehicle body along the vertical direction in this case.

Variations in the plots which can be referred to offset between curves are due to the positioning of the optical trigger that starts recording data. Any other differences can be either due to non-uniformities or from dynamic effects during the tests.

5 FE Simulations of Testing

This chapter discusses the results from LS DYNA simulations performed to correlate the experimental tests.

5.1 FE Model of Test Rig

In the FE simulations the test setup was replicated to give a similar representation of the tests. Since the test rig was built to be sufficiently rigid, the weld connections were defined by applying constrains using nodal rigid bodies. The clamping of the test rig to the Fall Rig bench was simulated by securing the FE model between rigid plates with single point constraints (SPC) applied around the boundary nodes securing all six degrees of freedom. The entire test rig was modelled in 4 node shell element with under-integrated element formulation. The impactor was modelled using 8 node solid elements with under-integrated element formulation and rigid material model. The impactor was constrained to have translation motion only in vertical direction. A beam element was created at the top of the impactor to replicate the bolt welded to the impactor in the tests. The bolt helps to secure the impactor to the hanger carrying the drop weight. The drop weight was modelled using element node mass added to the end node of the beam element. The end nodes were also used to measure the acceleration signals in the simulations and axial forces in the beam element.

The figure 5.1 and 5.2 show the FE models of the test rigs used in LS DYNA simulations to correlate the physical tests performed in the drop tower.

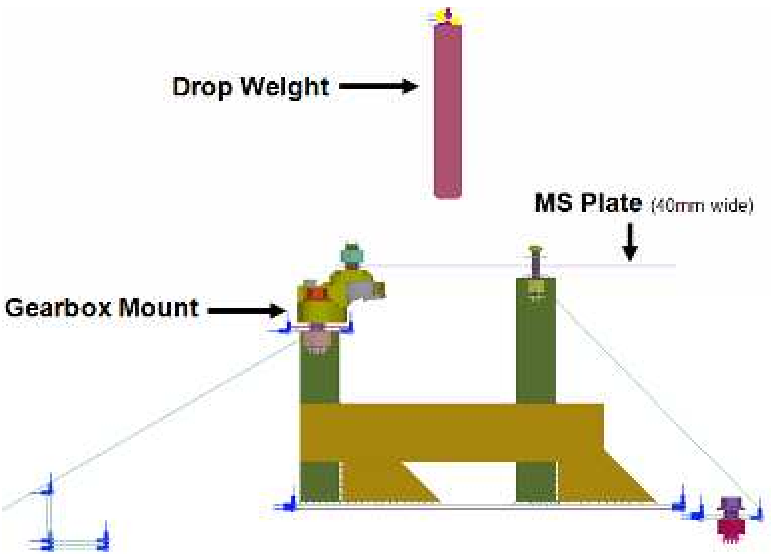

Figure 5.1 FE model for Test Rig for Gearbox Mount physical testing.

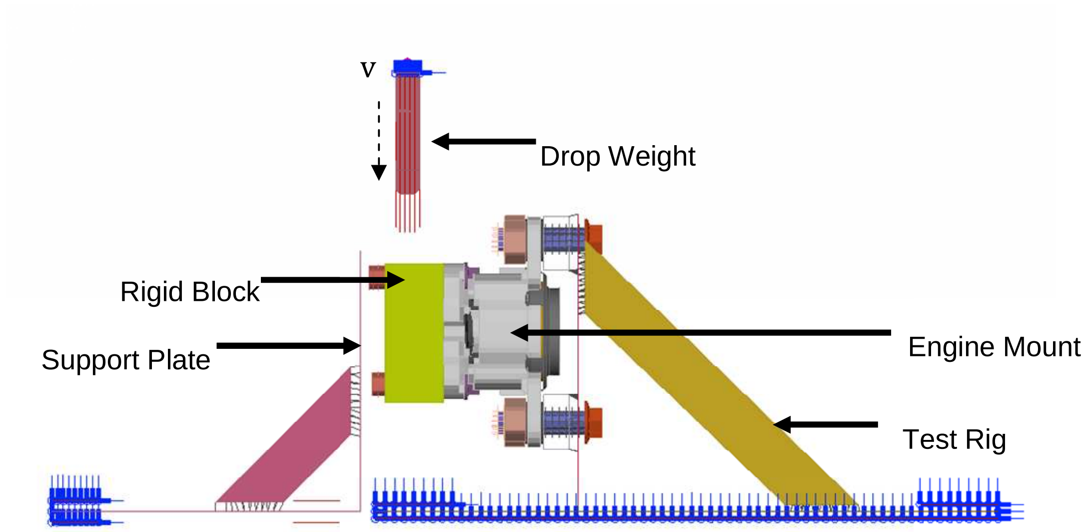

Figure 5.2 FE model for Test Rig for Engine Mount physical testing.

5.2 Gearbox Mount Test Simulations

For the test in phase one with lower gearbox mount, the correlation was performed with two FE models, one with hex mesh and another with tetra mesh for the lower gearbox mount. The objective of doing so was to check if the results are consistent with both the types of mesh and if the material card being used for aluminium silicon (AlSi12 Fe) casting is mesh dependent.

The figure 5.3 shows comparison of time sequence for crack propagation in hex and tetra meshed models:

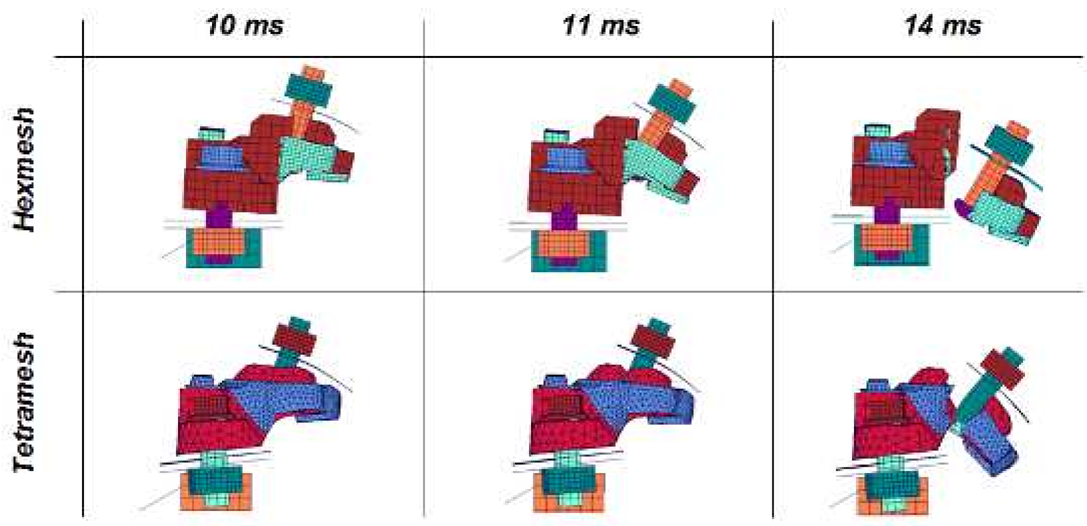

Figure 5.3 Time Sequence for crack propagation in hex and tetra meshed models

In figure 5.3 it can be observed that the crack begins in the hex mesh earlier than in tetra mesh. The crack propagates in 4 ms in hex mesh, 3 ms in tetra mesh and 2 ms in actual test. The tetra mesh is more realistic in capturing the crack propagation. The crack propagation is mesh dependent.

Figure 5.4 – 5.6 show the Force Vs Time plots comparing the FE simulations of hex and tetra mesh for gearbox mount with the results from the physical tests performed in phase one.

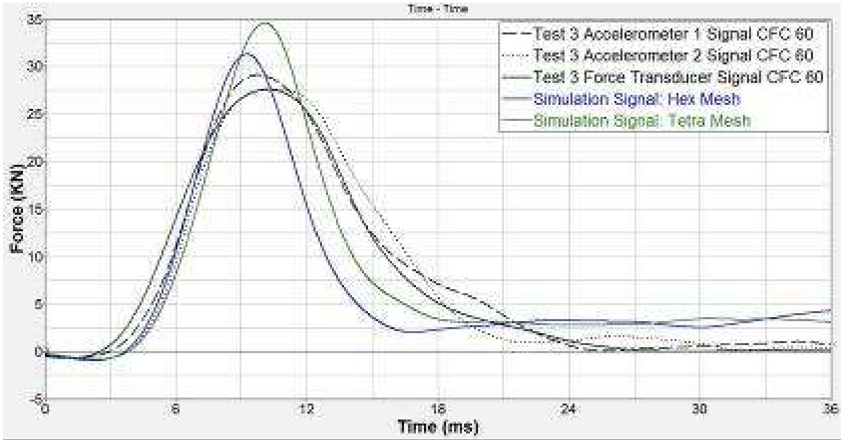

Figure 5.4 Force Vs Time phase 1 Test 03

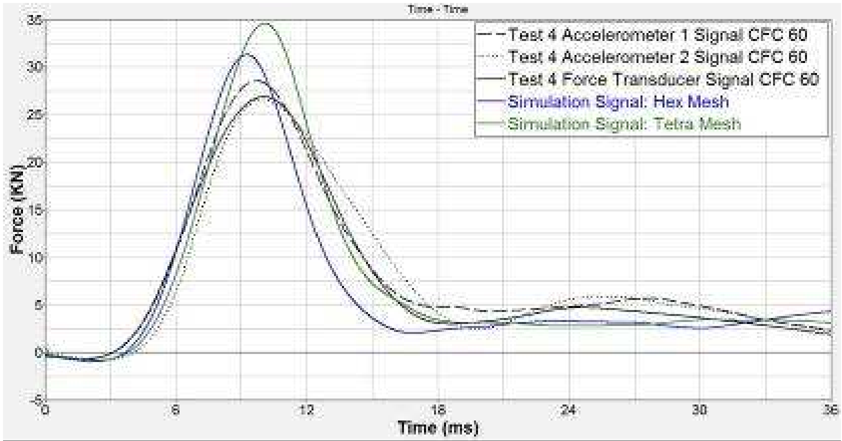

Figure 5.5 Force Vs Time phase 1 Test 04

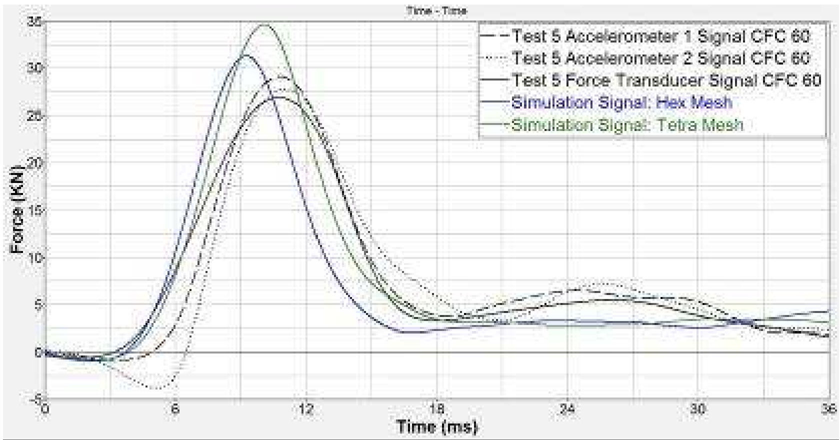

Figure 5.6 Force Vs Time phase 1 Test 05

5.2.1 Analysis & Discussion

The simulations force curves follow a similar trend to the test results from gearbox mount testing. The peak force is obtained to be around 31kN and 35kN for hex and tetra meshes respectively. These values are 12% and 25% higher compared to the experimental peak force of 28kN. Approximate friction value is taken into consideration between the mild steel plate and the clamping plates on the test rig because it is very difficult to predict the friction between them.

The material model being used to define AlSi12 Fe casting is tuned to cause fracture in the gearbox mount. Details on material model tuning can be found in APPENDIX B.

5.3 Simulation Results for Engine Mount Testing

5.3.1 Test with Hydraulic Fluid in Engine Mount

The figure 5.7 shows visual comparison between the physical test and LS DYNA simulations of the engine mount response to dynamic loading. The engine mount contains hydraulic fluid.

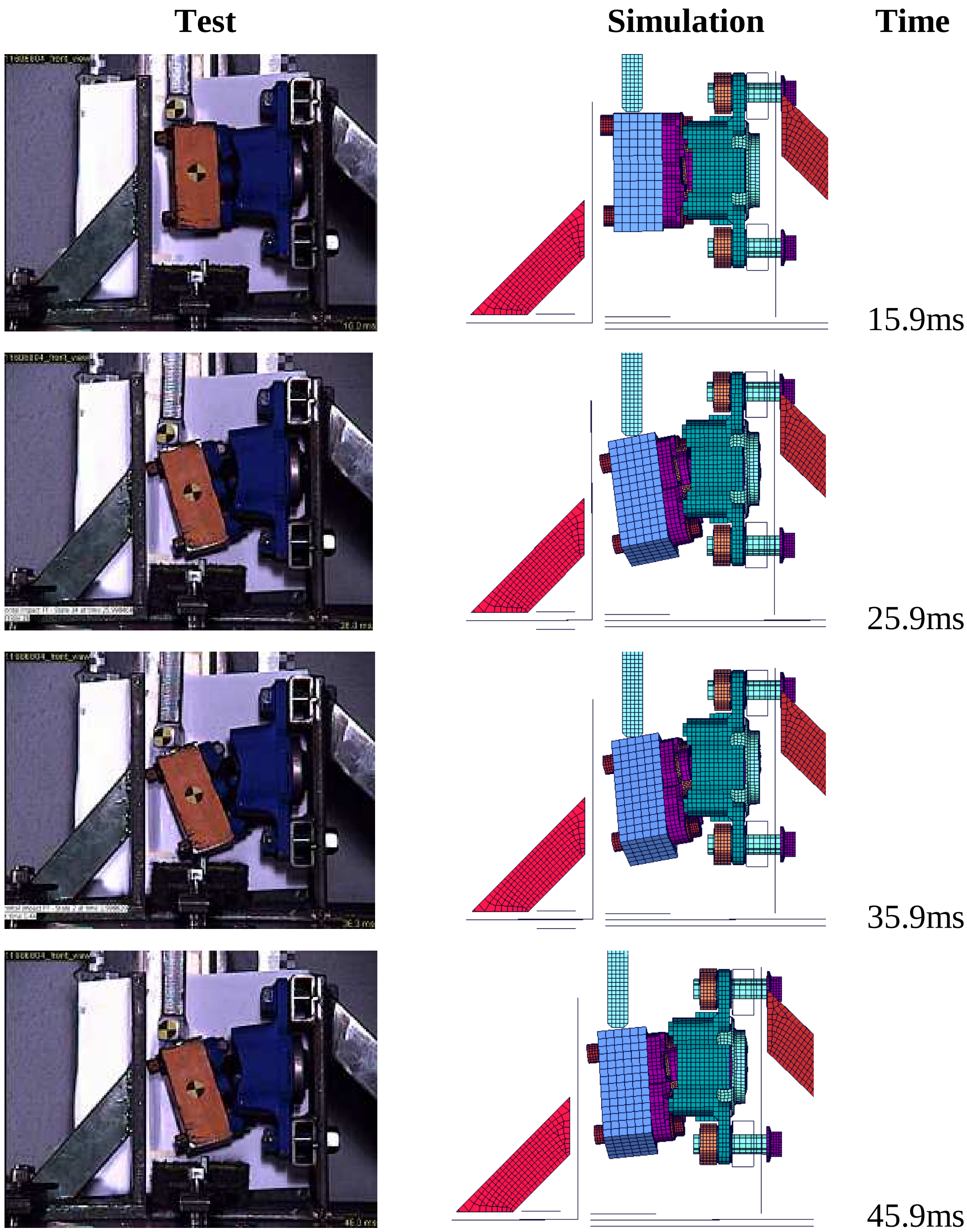

Figure 5.7 Time Sequence in physical tests and LS DYNA simulation on engine mounts with hydraulic fluid.

Figures 5.8 – 5.10 shows the comparison of the results between FE simulation and the physical test on the engine mounts containing hydraulic fluid.

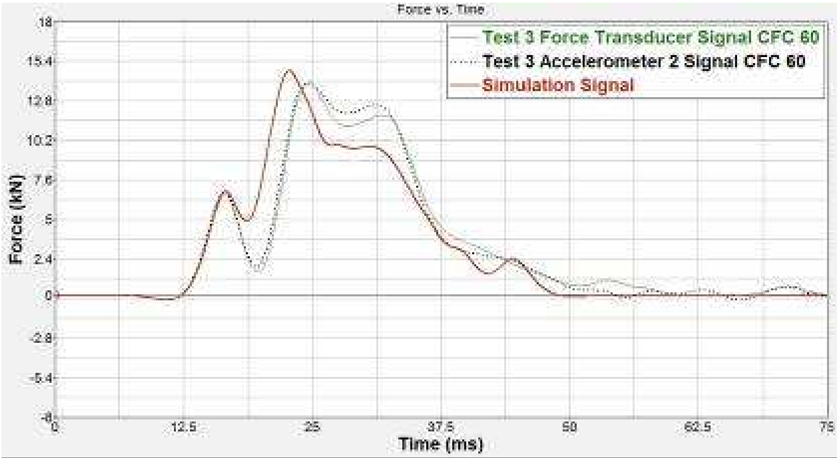

Figure 5.8 Force Vs Time phase 2 Test 03

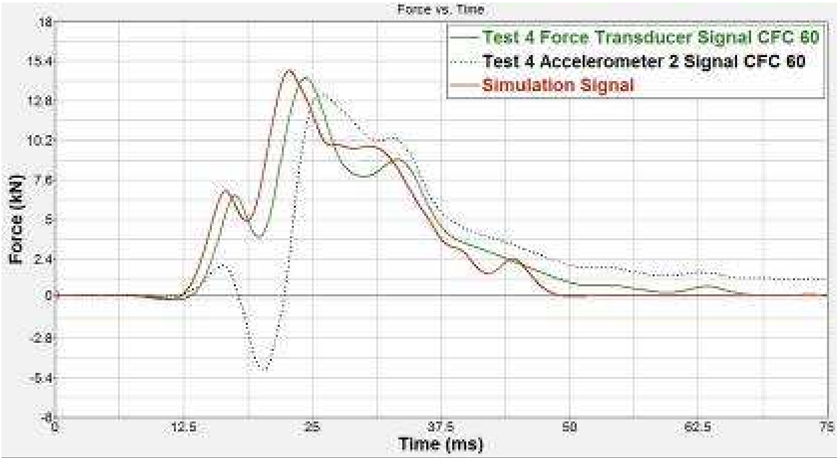

Figure 5.9 Force Vs Time phase 2 Test 04

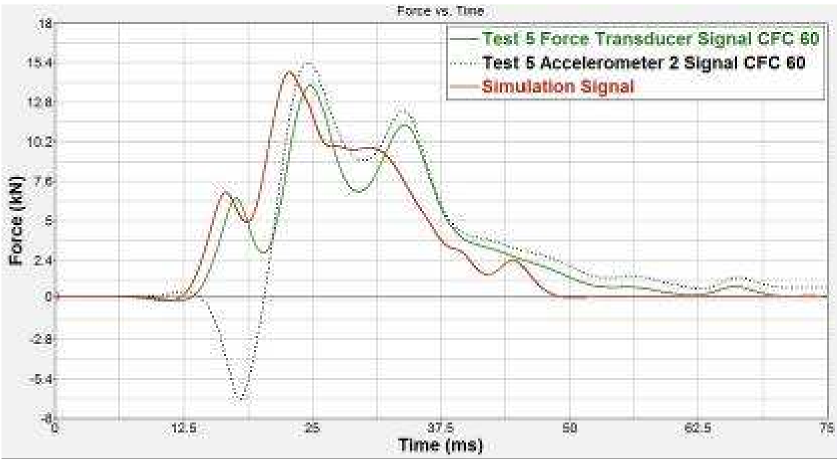

Figure 5.10 Force Vs Time phase 2 Test 05

5.3.2 Test without Hydraulic Fluid in Engine Mount

The figure 5.11 shows visual comparison between the physical test and LS DYNA simulations of the engine mount response to dynamic loading. The engine mount does not contain hydraulic fluid.

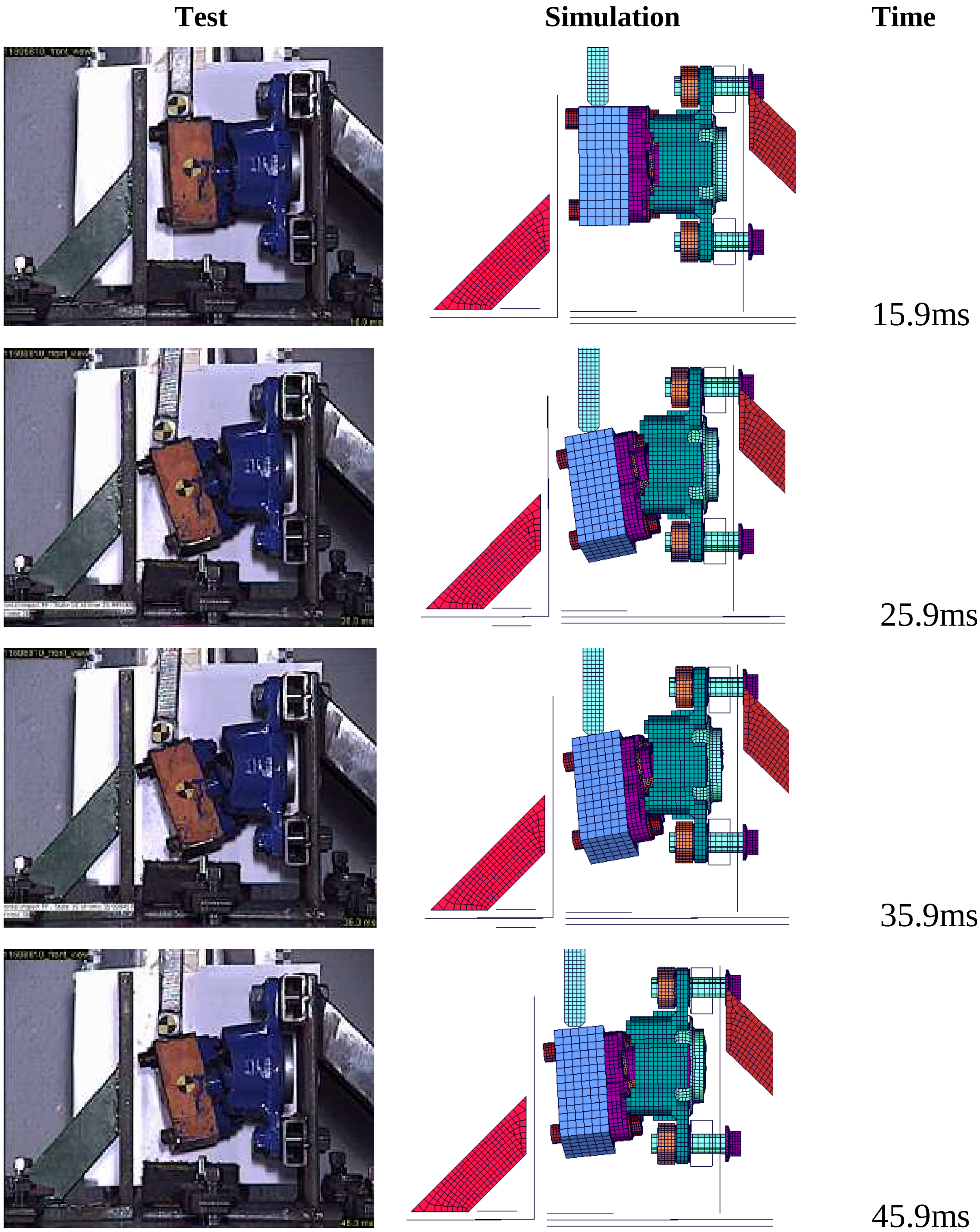

Figure 5.11 Time sequence for physical tests and LS DYNA simulation on engine mount without the hydraulic fluid.

Figures 5.12 – 5.13 shows the comparison of the results between FE simulation and the physical test on the engine mounts containing hydraulic fluid.

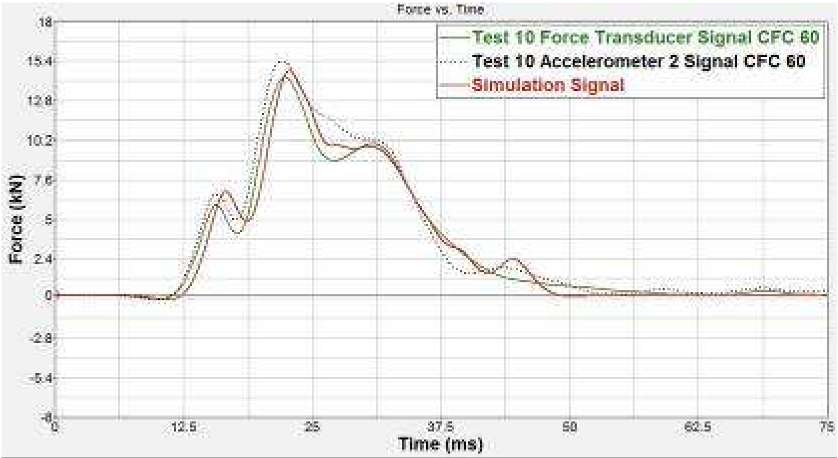

Figure 5.12 Force Vs Time phase 2 Test 10.

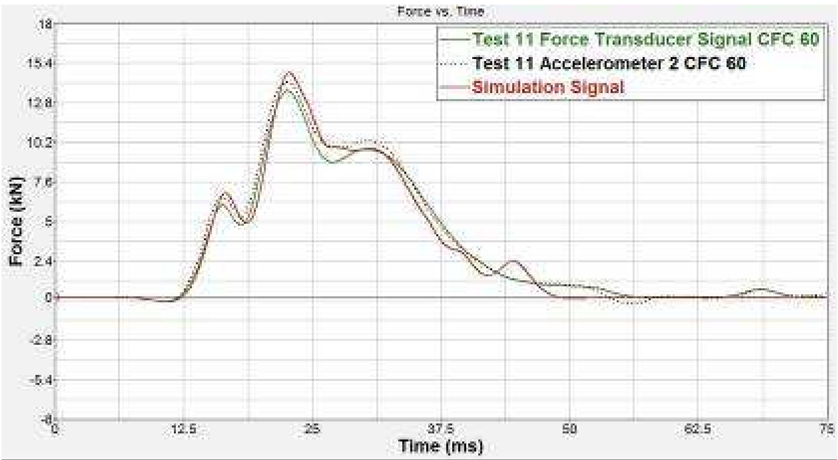

Figure 5.13 Force Vs Time phase 2 Test 11.

5.3.3 Analysis & Discussion

The material model for the aluminium silicon casting was tuned in the correlations for gearbox mount testing and was applied in performing LS-DYNA simulations for complete engine mount tests. Figures 5.4 and 5.11 shows the time sequence between the simulations and the physical test to evaluate the behaviour of the rubber bushing in the engine mounts and also to validate the effect of the hydraulic fluid. Until 26ms the rigid block in the simulations exactly follows the same path as in physical tests. Then the rotation is more in the simulation. This can be due to the fact that the foam under the rigid block was not modelled since it was only used to stop the fluid flow in case of leakage. Another factor that has been approximated is the gravity acting on the lower gearbox mount due to addition of the rigid block which can cause rotation of the block. An important aspect that needs to be addressed is the friction between the support plate and the bolt on rigid block which is difficult to measure. Even though the surface was lubricated, there was significant friction still acting on these parts and based on engineering judgement an approximate value is chosen. Although many approximations have been made the rubber bushing behaves very well and also the simulation graphs have similar trend to physical tests.

6 Validation of New FE Model in Full Vehicle Simulations

The FE model for the LHS engine suspension mount validated from the drop tests in the Fall Rig was incorporated into the full vehicle FE model for Volvo V60 vehicle by replacing the existing FE model. The complete vehicle FE model was then used to simulate the 35 mph full frontal rigid barrier impact test and the 40 mph offset deformable barrier impact test.

The performance of the new FE model for the engine mount was evaluated by comparing the time at which the crack develops in the lower gearbox mount during the crash and location of the crack with respect to the observations made from the engine mounts obtained from the actual full vehicle crash tests.

The 35 mph full frontal rigid barrier impact test is a regulatory test implemented by U.S National Highway Traffic and Safety Administration (NHTSA). The test involves the vehicle being propelled into a rigid barrier at a speed of 35 mph with full width overlap. The barrier is stationary while the vehicle is run into it. The figure 6.1 shows the top view of the test setup.

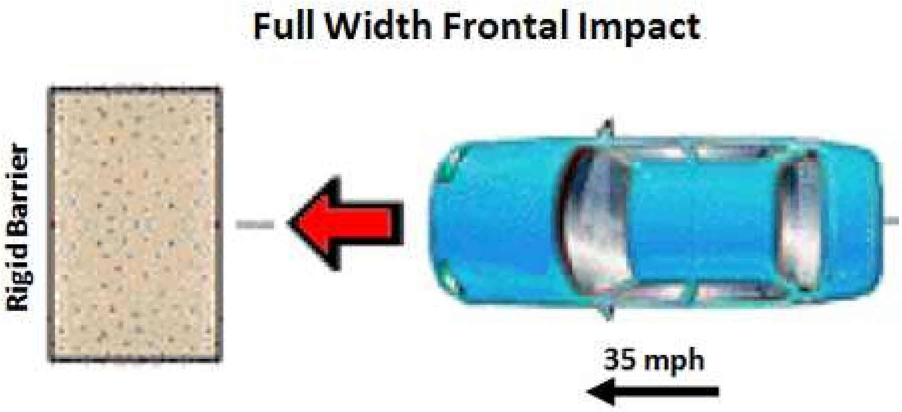

Figure 6.1 35 mph Full Frontal Rigid Barrier Impact Test

The offset deformable barrier impact test was developed by European Enhanced Vehicle-safety Committee as basis for legislation (where the initial velocity is 35mph). Each car tested is subjected to an offset impact into an immovable block fitted with a deformable aluminium honeycomb face. This impact is intended to represent the most frequent type of road crash, resulting in serious or fatal injury. It simulates one car having a frontal impact with another car of similar mass. As most frontal crashes involve only part of the car's front, the test is offset to replicate a half width impact between the cars. In the test, this is replicated by having 40 percent of the car impact the barrier.

The figure 6.2 shows the top view of the test setup:

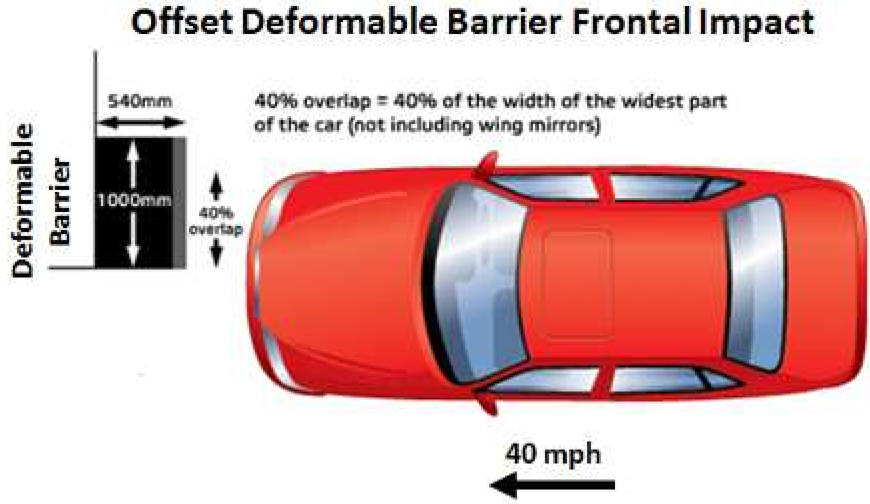

Figure 6.2 40 mph Offset Deformable Barrier Frontal Impact Test

6.1 Results

Figures 6.3 – 6.8 show the time sequence during the full vehicle crash simulations with existing and the new FE model for the LHS engine suspension model.

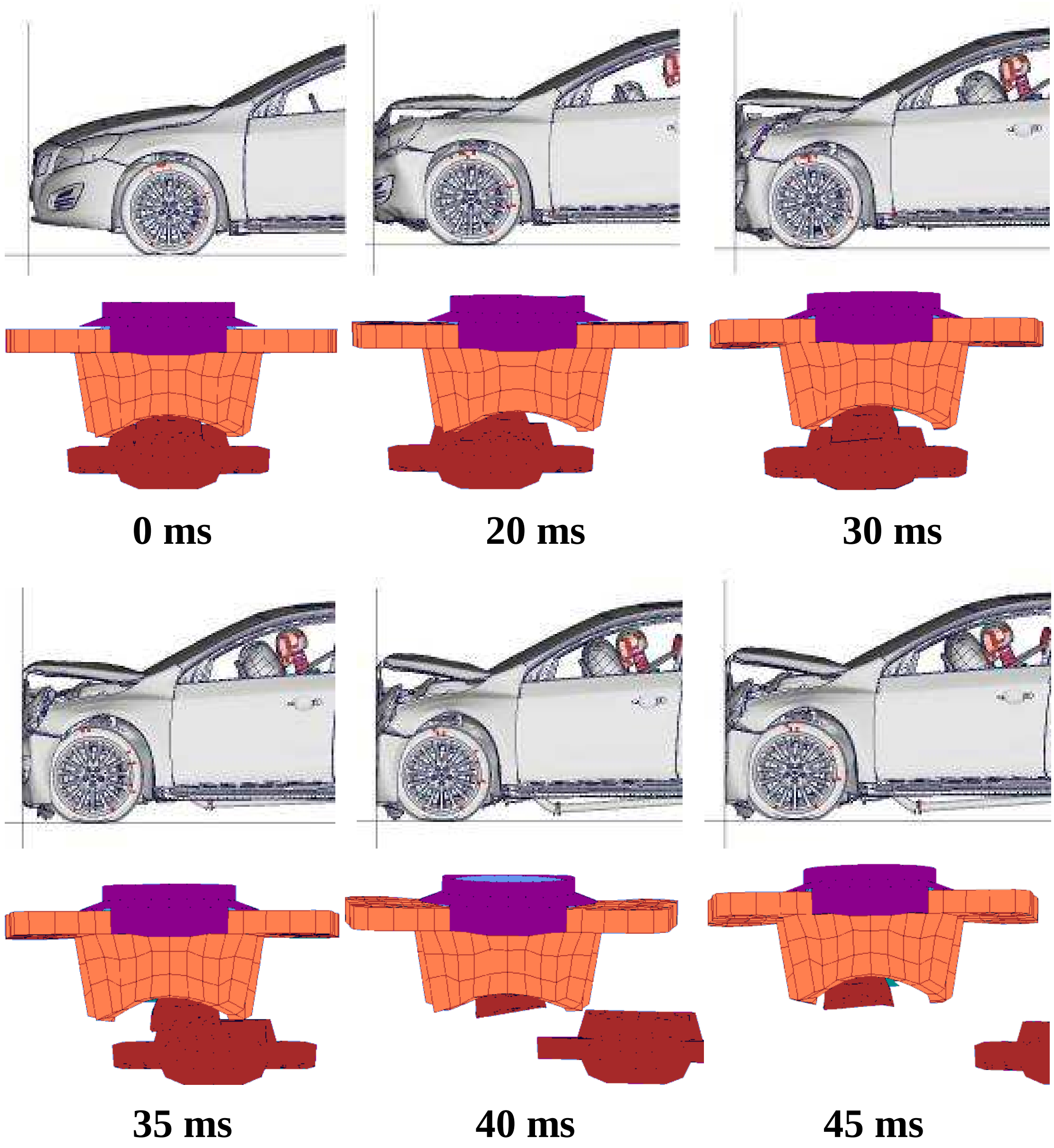

Figure 6.3: Time sequence in a 35 mph Full Width Frontal Impact Crash Simulation with existing FE model for LHS Engine Suspension Mount.

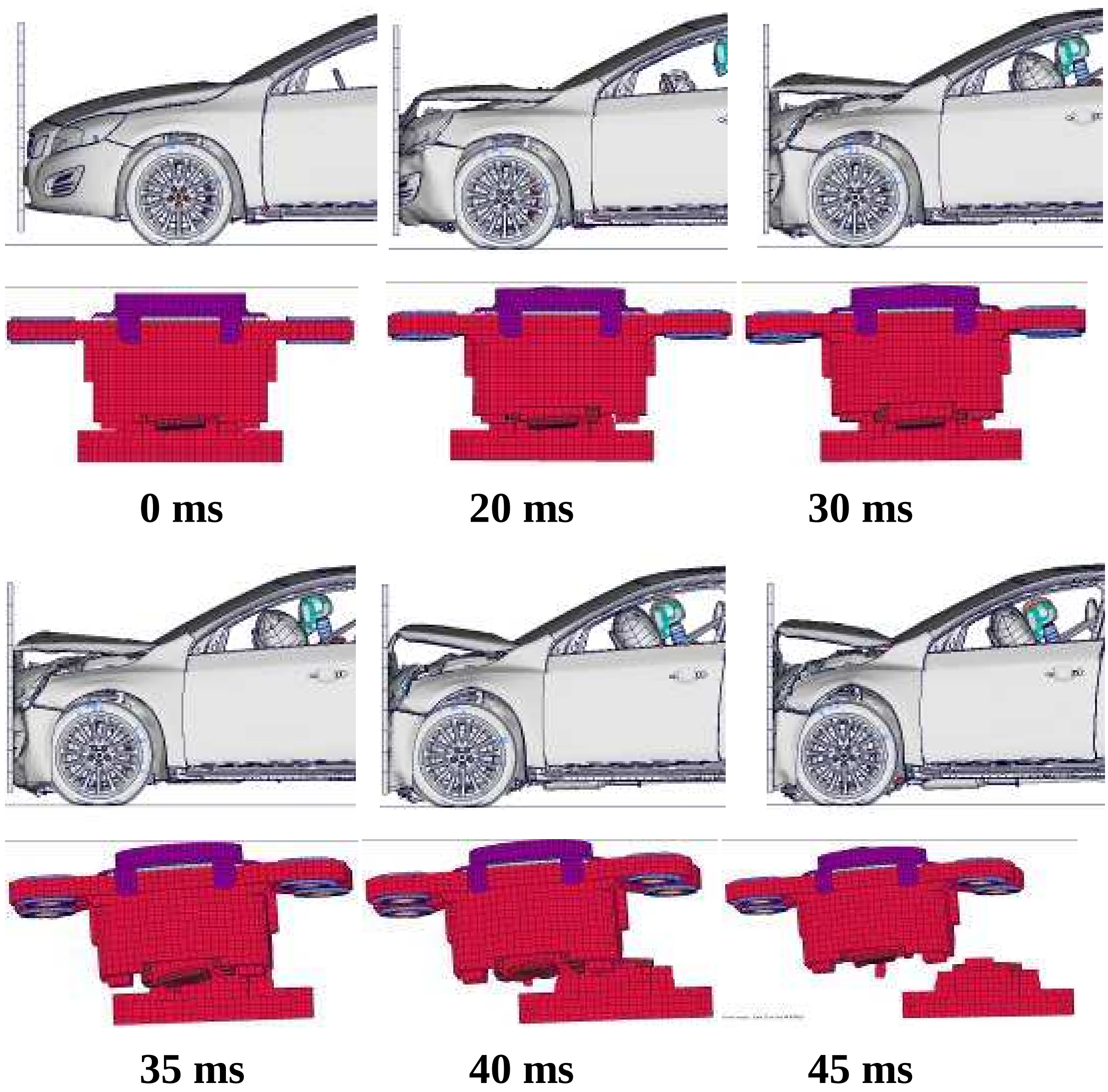

Figure 6.4: Time sequence in a 35 mph Full Width Frontal Impact Crash Simulation by incorporating improved FE model for LHS Engine Suspension Mount with hex mesh for lower gearbox mount bracket.

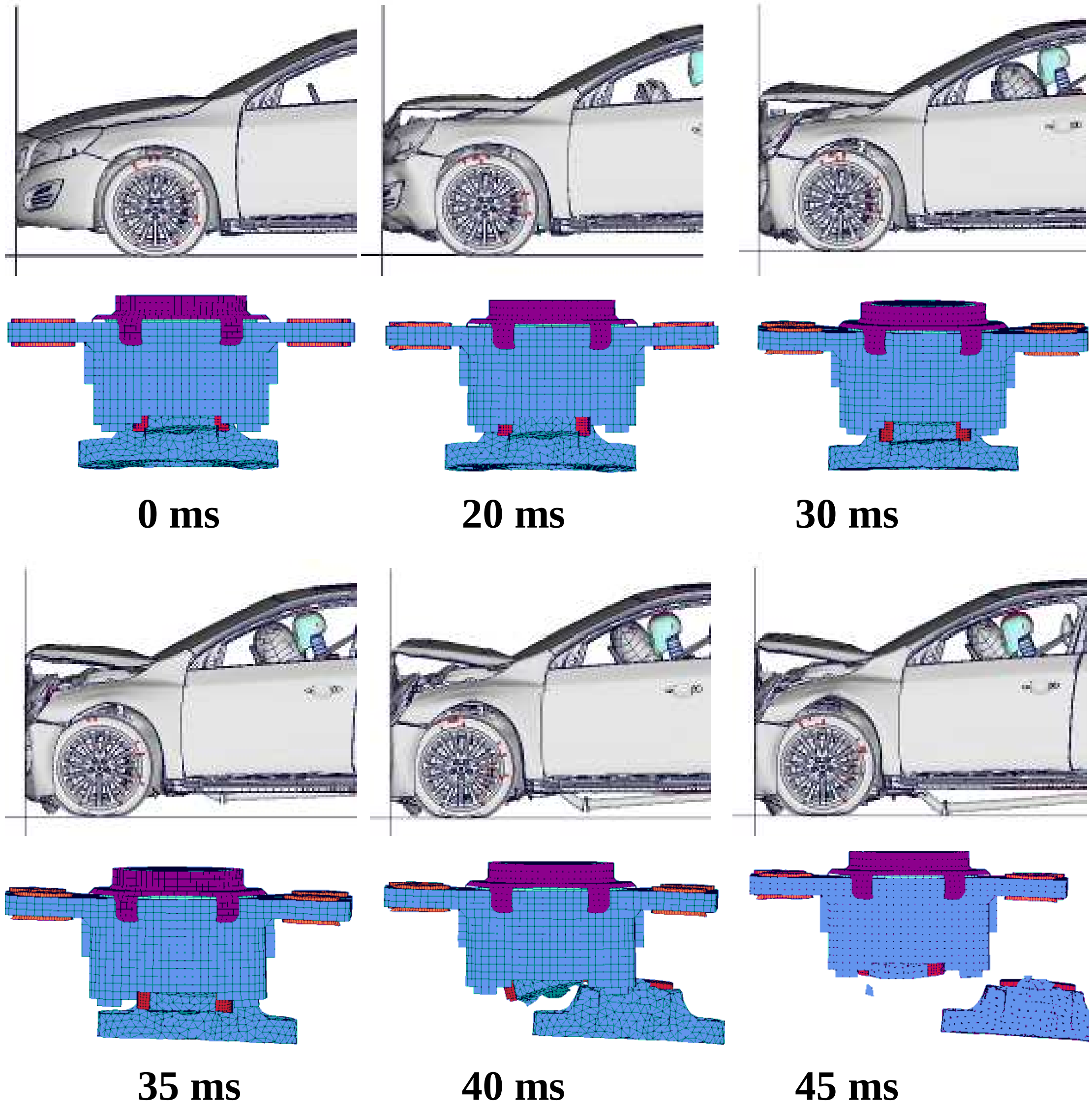

Figure 6.5: Time sequence in a 35 mph Full Width Frontal Impact Crash Simulation by incorporating improved FE model for LHS Engine Suspension Mount with tetra mesh for lower gearbox mount bracket.

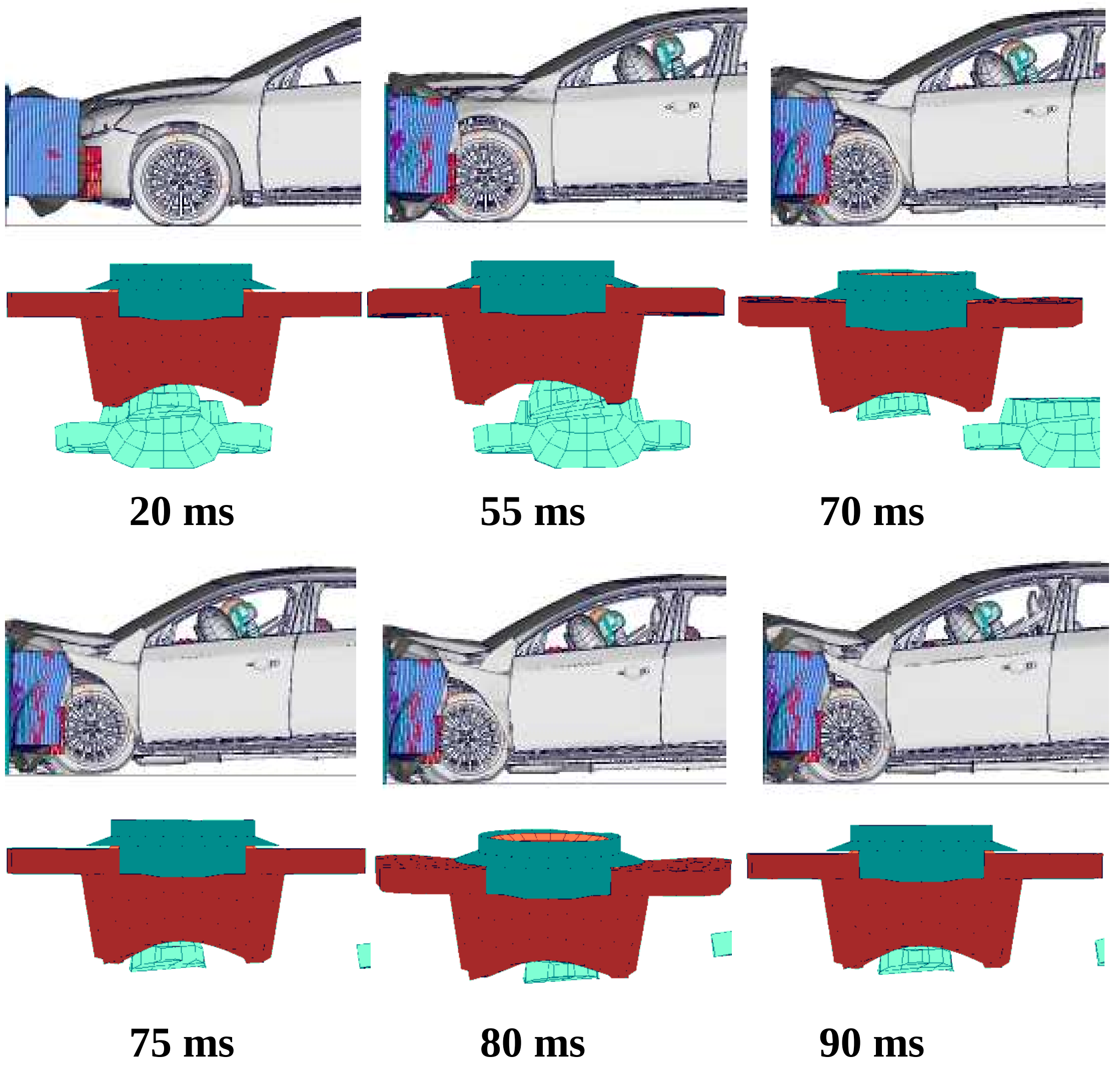

Figure 6.6: Time sequence in a 40 mph Offset Deformable Barrier Frontal Impact Crash Simulation with existing FE model for LHS Engine Suspension Mount.

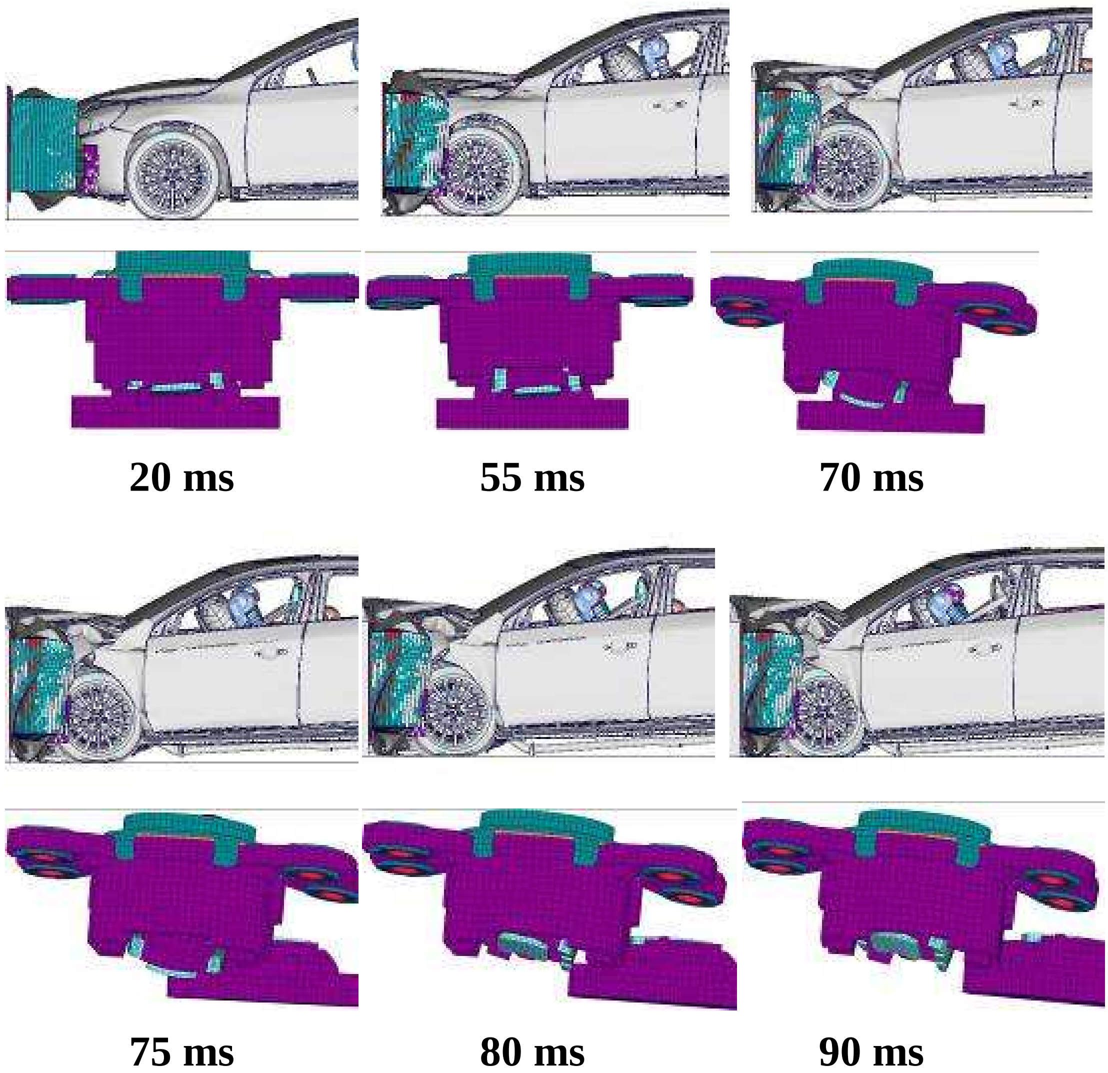

Figure 6.7: Time sequence in a 40 mph Offset Deformable Barrier Frontal Impact Crash Simulation by incorporating improved FE model for LHS Engine Suspension Mount with hex mesh for lower gearbox mount bracket.

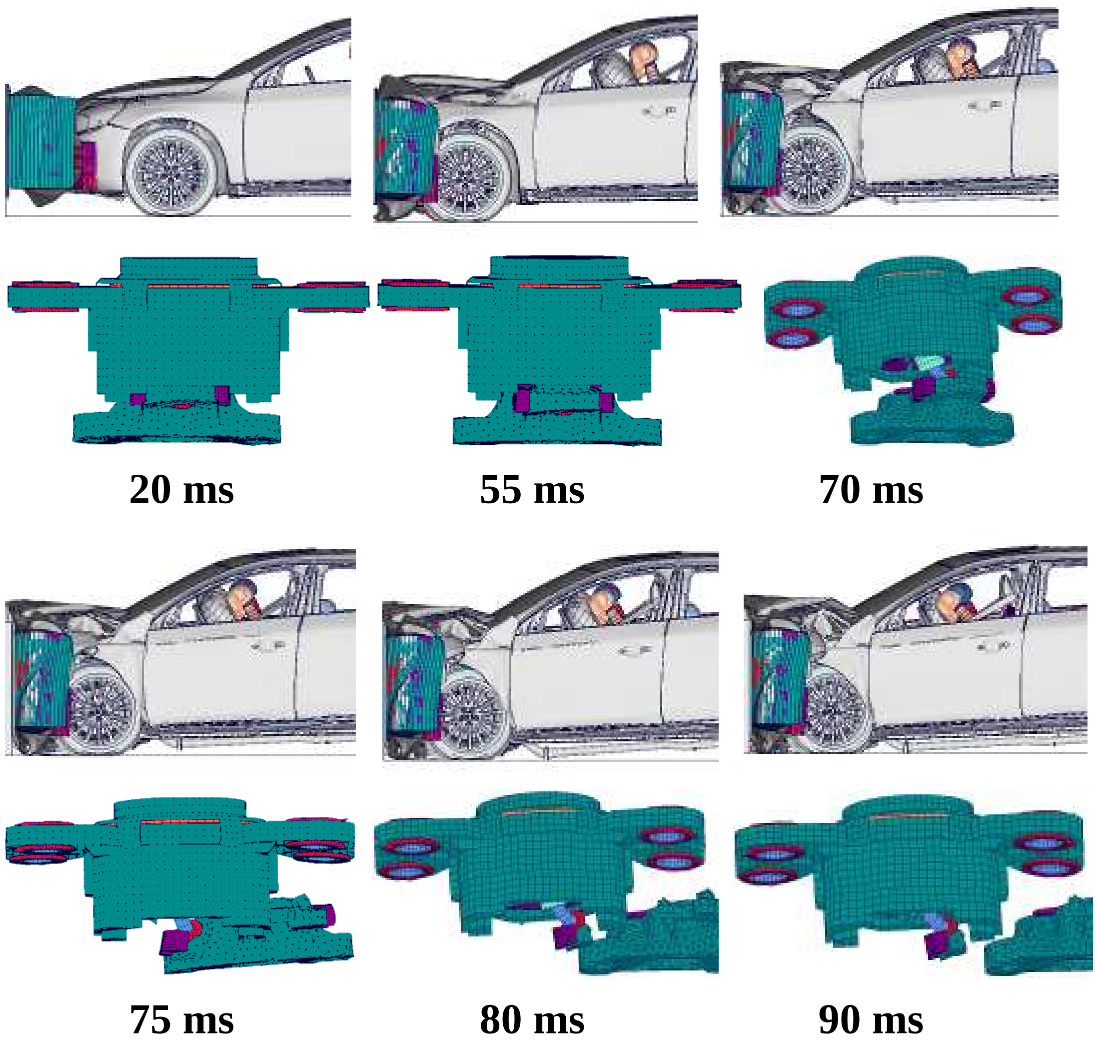

Figure 6.8: Time sequence in a 40 mph Offset Deformable Barrier Frontal Impact Crash Simulation by incorporating improved FE model for LHS Engine Suspension Mount with tetra mesh for lower gearbox mount bracket.

6.2 Analysis of Results

The tuned material card for aluminium silicon (AlSi12 Fe) castings and the new FE model for the LHS engine suspension mount were used in the full vehicle crash simulations. It was observed that the new FE model had good capabilities in capturing the dynamic movements in the rubber bushing of the engine mount. However, there was no failure caused in the aluminium silicon casting parts that was observed in the parts recovered from the actual crash tests. From the study of the parts recovered from actual crash tests, it was observed that there was a failure in the lower gearbox mount bracket in every test. Thus it was concluded that the material model for aluminium silicon casting was stiffer. The material model was then modified as explained in APPENDIX B.

Figure 6.3 – 6.5 compare the performance of the existing and the new FE model of the LHS engine suspension mount in a 35 mph full frontal rigid barrier impact simulation. In the new FE model, the deformations of the rubber bushing during the initial phase in the crash due to inertia of the engine and the gearbox causing the gearbox mount to move forward causing bending moment and compression in the rubber bushing due to load transfer though the screw is well captured. The same motion in the existing FE model was observed to be more pronounced in magnitude. It was concluded that this might be due to the use of multidimensional spring elements instead of a detailed physical model of the screw and the rubber bushing resulting in less lateral stiffness in the existing FE model of the engine mount. The failure in the engine mount occurs in the lower gearbox mount aluminium casting once the engine and the gearbox are being pushed rearwards towards the firewall. Thus the engine and gearbox are detached from the body side members approximately 35 ms into the crash. This value is same as observed with use of existing FE model for engine mounts. The location of the crack in the new FE model of gearbox mount is similar to the one seen in the engine mount recovered from full vehicle crash test. It is important to note that location of the crack on the aluminium silicon casting can be more accurate with a finer mesh. However, to avoid mass scaling problem the element length for the mesh representing aluminium silicon casting was kept approximately 5.5 mm. The performance with hex and the tetra mesh for the lower gearbox mount are similar. Thus the new FE model is reliable to predict the behaviour of the engine suspension mount in 35 mph full frontal impact into a rigid barrier.

Figure 6.6 – 6.8 compare the performance of the existing and the new FE model of the LHS engine suspension mount in a 40 mph offset deformable barrier frontal impact simulation. The behaviour of the engine mount in the rubber bushing during the initial phase of the crash was well captured. The failure in the lower gearbox mount occurs 70 ms into the crash. Thus there is a lag in the time when engine and the gearbox are detached from the body side members with new FE model as compared to the existing FE model. It is important to note that the late failure in the engine mount cannot be considered to be inaccurate since there is no data available from the actual full vehicle crash tests about the exact time of failure in the engine mount. However, it will be necessary to study the effect of late failure in the engine mount on the crash pulse in the passenger compartment before considering the new FE model to be reliable for use in simulations for this test. This verification was not performed in the thesis work due to time constraints. The results are similar for new FE model with hex as well as tetra mesh for the lower gearbox mount.

7 Conclusions

The work in this mater thesis was concentrated in developing a detailed physical FE model for the LHS engine suspension mount existing in Volvo S60 / V60 vehicle to be used in full vehicle crash simulations. The important features that the FE model needs to capture in a crash simulation are dynamic behaviour of the rubber bushing and failure in the aluminium silicon casting.

The rubber was modelled with tetra elements with critical element length of 4.5 mm. It was defined by using hyper elastic material that uses Yeoh material model. More information on the material modelling of rubber for the bushing is available in Appendix B and [9].

The aluminium silicon casting (AlSi12 Fe) was modelled using hex and tetra elements with minimum element characteristic length of 5.5 mm. A MF GenYld + CrachFEM material model was used for cast part that is capable of predicting the elasto–plastic behaviour and crack failure. The material model was tuned based on the physical tests performed on the engine suspension mount in the Fall Rig and the observations made on the engine mounts recovered from the actual crash tests. The material models had to be tuned by modifying the failure strain curves to get fracture in the engine mount casting. Details for this material model are available in APPENDIX B.

The new FE model was validated in full vehicle crash simulations. The results from these simulations show that the new FE model is capable in predicting the behaviour in its rubber bushing and crack in the aluminium silicon casting. The failure in the engine mount occurs 35 ms in case of 35 mph full frontal rigid barrier impact test and 75 ms in case of 40 mph offset deformable barrier test. However, additional data from the actual crash test will be necessary to validate the exact time at which the engine mount fail detaching the engine and gearbox from the body side members. Material testing for AlSi12 Fe material must be performed to obtain data on failure strains under dynamic loading and validating the GenYld + CrachFEM material model that has been tuned by following a trivial method in this master thesis. The performance of the engine mount with hex as well as tetra mesh was observed to be consistent.

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APPENDIX A

Introduction to Explicit Finite Element Analysis

Direct integration solves the dynamic equation of motion using the difference expression by time integration algorithms that are associated with the geometric and material nonlinearities. Dynamic analysis is necessary in an engineering problem, if there are any mass inertial effects and time dependent functions. Crash simulations are related to impact loading that creates wave propagation through the elements, causing large deformations in the vehicle. Crash simulation problems are highly nonlinear and require dynamic analysis. Therefore, explicit direct integration method is best suited for this purpose.

For a system of nonlinear characteristics, the equation of motion is a function of inertial, elastic, damping and the external force.

$$M\ddot{U} + \ C\dot{U} + KU = F \tag{1}$$

Where M, C and K are mass, damping and stiffness matrices, and U is the nodal displacement. It is assumed that the mass and stiffness matrices are positive definite for the structure to have rigid-body motion as a part of its response.

A modified central difference time integration method based on velocity and acceleration is applied in LS-DYNA.

$$\dot{U}_t = \frac{U_{t+\Delta t} - U_{t-\Delta t}}{2\Delta t} \tag{2}$$

$$\ddot{U}_t = \frac{U_{t-\Delta} - 2U_t + U_{t+\Delta t}}{\Delta t^2} \tag{3}$$

The above equations (2.2) and (2.3) are substituted in the equation of motion which is a function of displacement.

$$(a_0 M + a_1 C)U_{t+\Delta t} = F_t - (K - a_2 M)U_t - (a_0 M - a_1 C)U_{t-\Delta t} \tag{4}$$

A method is said to be explicit, if the mass and damper matrices are diagonal, and then it is not required to solve the equations. Otherwise it is called an implicit method. But then the damping terms should be evaluated in the previous time increment.

The external force is a function of nodal displacements because they depend on the structure. When pressure forces are applied to the structure, they undergo large deformations. The internal energy of a system is also a function of nodal displacements since it is dependent on the stress which in turn depends on the strain (Belytschko et al.) [1].

Initial conditions are set in equation (1) to determine the internal and the external forces on each element. These forces are substituted in the equation of motion to get the acceleration at time$t - \frac{\Delta t}{2}$. Now the velocity and the displacements are updated as the time increments. The updated nodal velocities and displacement are put in the strain displacement equations to get new internal and external forces. Hence it is a continuous process, where the nodal velocities and displacements are updated with time increment outputting stress and strains in the structure. In the equation (1) we see that there are damping terms, resulting in system lag by half time step. This means that the velocities are calculated from previous time step is added to the velocity of the current time step by integrating the acceleration with time. The velocities that result are an integral of acceleration with respect to time using the central difference method. And again the velocity is integrated with time to get displacement.

Stability and Time Step

In an explicit method to achieve accurate result requires smaller time increment. The above equation (4), is said to be conditionally stable, that is if the time step exceeds the critical value it would result in a wrong solution. To determine the time step, the velocity and acceleration (2) and (3) obtained from the central difference method at time 't', are inserted into the equation of motion (1).

Therefore for an undamped equation of motion the required time step is and equal to the critical time step.

$$\Delta t \leq \frac{2}{\omega_{max}} \tag{5}$$

Where ω_{max} is the highest natural frequency of the system.

The critical time step depends on the material properties and the size of the element in explicit time integration.

$$\Delta t_{crit} = min \frac{l_e}{c_e} \tag{6}$$

Where l_e and c_e are the minimum characteristic length of the element and speed of sound in the element material respectively. Which is also called the courant condition in finite difference methods. LS-DYNA calculates the time step for each element and the minimum element time step is used in the simulation.

From the equation (6), it is known that the time step is dependent on the material properties the speed of sound in an eight node solid element of that material is;

$$c_e = \sqrt{\frac{E}{\rho(1-v^2)}} \tag{7}$$

Where E, ρ and v are the Young's modulus, density and poisson's ratio for that material.

Mass Scaling

Since very small time steps are needed for an explicit analysis the element size is of greater importance for numerical stability. LS-DYNA automatically detects elements which have time steps less than the critical time step and adds nonphysical mass to its nodes in order to achieve numerical stability at higher time steps. For the simulation in LS-DYNA, automatic mass scaling is used, controlled by DT2MS in the *CONTROL_TIMESTEP [8].

APPENDIX B

Material Model Description for Rubber and Aluminium Silicon Alloy

Rubber Material Model (Hyperelastic Yeoh Model)

The rubber bushing in the engine suspension mount is modelled using material card 77, *MAT_HYPERELASTIC_RUBBER. Yeoh model as hyperelastic rubber material model has been used to develop this material card. This section describes the Yeoh material model.

The Yeoh model in third order reduced polynomial form describes isotropic incompressible rubber like material. The strain energy density function W, implemented in FE programs capable of handling hyperelastic materials is given by following equation:

$$W(I_1, I_2, I_3) = \Sigma_{\mathrm{i,j,k=0}}^{\infty} \mathrm{C_{ijk}} (I_1 - 3)^i (I_2 - 3)^j (I_3 - 1)^k \tag{8}$$

In case of total incompressibility $I_3 = 0$, above equation changes to following;

$$W(I_1, I_2) = \Sigma_{\mathrm{i,j=0}}^{\infty} \mathrm{C_{ij}} (I_1 - 3)^i (I_2 - 3)^j \tag{9}$$

Where C_{ij} are unknown constants. The above equation in expanded form neglecting higher order terms and neglecting the terms that include I_2 due to low dependency on the second invariant for carbon black filled natural rubbers is written as follows;

$$W(I_1) = C_{10}(I_1 - 3) + C_{20}(I_2 - 3)^2 + C_{30}(I_3 - 3)^3 \tag{10}$$

The rubber material parameter C_{ij} can be approximately determined from its initial shear modulus. The approximate relations are given by following equations;

$$C_{10} = \frac{G}{2};\ C_{20} = \frac{G}{20};\ C_{30} = \frac{G}{200} \tag{11}$$

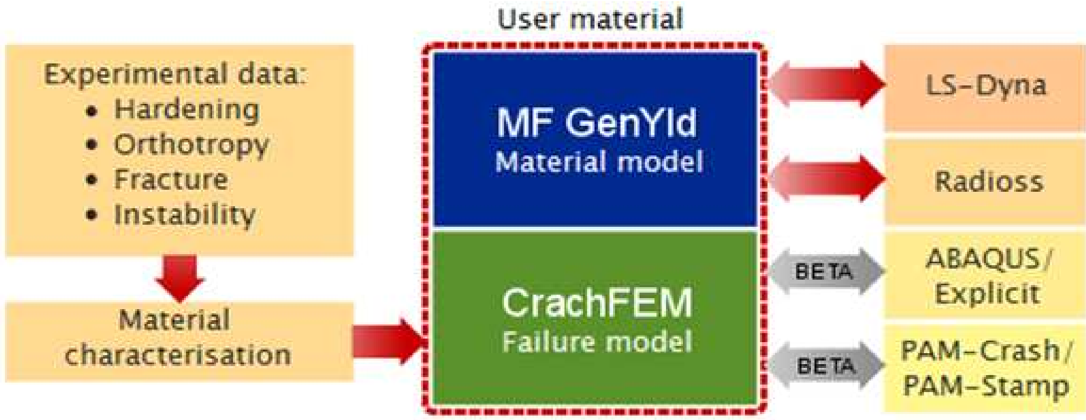
It is important to note that the parameter C_{10} influences the behaviour in lower strain rates and parameter C_{30} in higher strain rates.

The material model for rubber bushing used in this thesis work has the following values: $C_{10} = 0.55$ MPa, $C_{20} = -0.05$ MPa and $C_{30} = 0.95$ MPa. More details for the rubber material model are found in [9].

Aluminium Silicon Material Model (MF GenYld + CrachFEM)

The information in this section is based from [13].

MF GenYld + CrachFEM is a comprehensive material and failure model that can be used in combination with commercial explicit FE code such as LS DYNA. It allows the characterisation of various metallic materials in LS DYNA for shell and solid element models. This approach of a universal material model allows the use of standardised experimental and theoretical methods for the characterisation of the material for FE analysis. The figure below shows the concept of interlinking experimental data with commercial FE codes through MF GenYld + CrachFEM material model.



Pictorial representation of interlinking the experimental data with FE codes through MF GenYld + CrachFEM (Reference: http://www.matfem.com/en/crachfem.html)

CrachFEM refers to the part of the material model that describes failure. It includes the fracture model and the prediction of local instability. The Mf GenYld part of the material model describes the elasto–plastic behaviour. The name refers to 'Generalised Yield Locus'.

The elasto–plastic material model MF GenYld (Generalised Yield Model) allows the combination of range of yield locus descriptions with a variety of strain rate dependent hardening models, including input of sampled curves. The isotropic hardening base models can be extended to include isotropic – kinematic hardening of metals and anisotropic, stress – state dependent hardening. The isotropic kinematic hardening model has the potential to correctly describe the materials that show a pronounced Bauschinger Effect. The anisotropic hardening is designed to model different hardening behaviour for tension, compression and shear.

The failure model CrachFEM comprises the evaluation of local necking by means of transient calculation of the forming limit strain for metals, of ductile normal fracture caused by coalescence of micro-voids and of ductile shear fracture due to shear band localisation. The criteria for ductile and shear fracture are consistent for shell and solid elements. The evaluation of ductile and shear fracture uses and integral approach in order to consider the element deformation history.

In order to predict the failure during crash simulation, the deformation and heat treatment history of the involved parts are taken into accounts.

The MF GenYld + CrachFEM material model for aluminium casting (AlSi12Fe) is modelled to predict two types of ductile fractures:

- Ductile normal fracture resulting from void coalescence
- Ductile shear fracture resulting from shear band localisation

Details for the ductile fracture laws are available in [13].

The MF GenYld + CrachFEM material is coupled to the FE code LS DYNA through user defined material model represented by card number 48 in LS DYNA material library. The fracture model curves are input into LS DYNA simulation through the user defined material model. The material model was tuned based on the physical testing performed in the Fall Rig and observations from the actual crash tests.

The procedure for tuning this material model can be summarized in the following points:

1. To increase the failure risk in the elements by decreasing the value of FRELIM below one but higher than zero (1>FRELIM>0) to initiate failure in the material at lower strains.
2. The value of FRELIM is increased to its ideal value of one and the values of fracture strains are decreased by modifying the fracture curves.

In the above discussion, FRELIM is a parameter defined in the user defined material card. It relates the failure risk for the element and controls element elimination by erosion. The ideal value is one at which there is elimination at 50 % probability of failure. A value of zero refers to no element elimination.

Decreasing the FRELIM value below one makes the material weaker in fracture. Thus the material fractures at lower force values and time required for crack propagation in smaller. Hence the material is weaker and brittle compared to its initial state.